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SHAFTING, PULLEYS, BELTING, ROPE TRANSMISSION AND SHAFT GOVERNORS

COMPILED AND WRITTEN

BY

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SHAFTING, PULLEYS, BELTING
AND ROPE TRANSMISSION

CONTENTS

INTRODUCTION

THIS handbook is intended to furnish the reader with practical help for the every-day handling of shafting, pulleys and belting. These are allied in the operation of plants and it is a pretty generally conceded fact that all three are much neglected by many operators.

A close perusal of these pages will enable the reader to determine the best course to pursue in the most common instances and in various troubles, and in all articles there are suggestions for similar cases which may arise.

For instance, the need of belt dressing as a preservative, now generally conceded by most authorities, is fully covered in Chapter XI and the result of a test made by disinterested parties to find the degree of efficiency of four of the best known dressings is given. The results are of importance to all belt users.

A portion of the book is also given to rope transmission which is in more general use to-day than ever before, and in this connection some advice is offered by experts as to the selection and care of the rope. Rope splices and how to make them will also prove valuable to many engineers.

The author wishes to make acknowledgment to various contributors to *Power* whose articles are used

herein, and to some special contributors, from whose articles small portions have been taken. Acknowledgment is also made to Stanley H. Moore, the author of "Mechanical Engineering and Machine Shop Practice" for the section on splicing.

HUBERT E. COLLINS.

NEW YORK, *November*, 1908.

I

SHAFTING HINTS [1]

In the installation, maintenance and repair of shafting, as in all other things, there is a right and a wrong way; and though the wrong way ranges in its defects from matters causing trivial inconvenience to absolute danger, the right too often — owing to lack of knowledge or discernment — finds but scant appreciation.

Where, as is often the case, the end of a shaft is journaled to admit of the use of an odd, small-bore pillow block or wall-box hanger, the journaled part should equal in length twice the length of the hanger bearing plus the length of the collar. The hanger can thus readily be slid out of the wall box, and the necessity of uncoupling this shaft length and removing it before access to the bearing for purposes of cleaning or repair is done away with.

A plank or board *A* (Fig. 1), about $\frac{1}{4}$ to $\frac{1}{2}$ inch longer than the distance from the bottom of the shaft to the floor, can be used to good advantage at such times to free the hanger of the shaft's weight, and to prevent the shaft's springing from its own weight and the pulleys it may be carrying.

Should it become necessary to place a pulley with

[1] Contributed to Power by Chas. Herrman.

half the hub on and half off the journaled part, this can readily be done by the use of a split bushing, as shown in sectional view of Fig. 1.

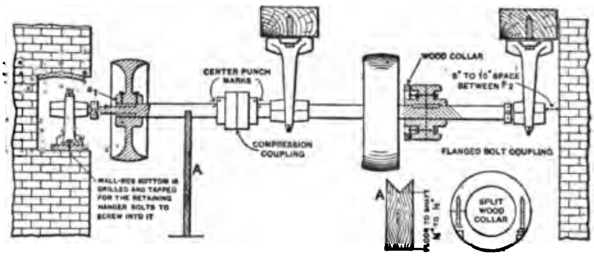


FIG. 1.

Very often a small-sized bearing is used and the shaft journaled off to act as a collar. Of this procedure it can only be said that if done with the idea of making a "good job" it signally fails of its object; if of necessity (a collar being insufficient), then the shaft is heavily overloaded and serious trouble will result, because of it.

It is advisable to center punch, or otherwise mark, the ends of both shafts held by a compression coupling close up against the coupling, and both edges of the coupling hub should have a punch mark just opposite and close to the shaft punch marks. These marks will serve at all times to show at a moment's glance any end or circumferential slippage of the shafts within the coupling. The same method can be resorted to for proof of pulley slippage.

When a new line of shafting is put up, the foot position of each hanger should be clearly marked out on their respective timbers *after* the shaft has been brought

into alinement. Hangers can thus be easily put back into their proper place should timber shrinkage or heavy strains cause them to shift out of line. This idea can be applied to good advantage on old lines also, but before marking out the hanger positions the shaft should be tried and brought into perfect alinement.

Hangers that do not allow of any vertical adjustment should not be used in old buildings that are liable to settle. Shafting so run pretty nearly always gets out and keeps out of level.

In flanged bolt couplings (Fig. 1) no part of the bolt should project beyond the flanges. And where a belt runs in close proximity to such a coupling, split wood collars should be used to cover in the exposed coupling flanges, bolt heads and nuts. Countershafts have been torn out of place times innumerable by belts getting caught and winding up on the main line.

Whenever possible a space of 8 to 10 inches should be left between the end of a shaft line and the wall. A solid pulley or a new coupling can thus readily be put on by simply uncoupling and pushing the two shaft lengths apart without taking either down. Ten inches does not represent the full scope of pulleys admissible, for so long as the pulley hub does not exceed a 10-inch length the pulley face (the more readily in proportion to the larger pulley diameter) can be edged in between the shafts.

Fig. 2 is an instance of bad judgment in locating the bearings. In one case this bearing overheated; the remedy is either to re-babbitt the old box or replace it with a new one.

Both pulleys were solid and the keys — headless ones — had been driven home to stay. The rims of both pulleys almost touched the wall, and the circumferential position on the shaft of both these pulleys was such as to preclude the possibility (owing to an arm of *a* being in a direct line with key B^1 and arm of *b* with key a^1) of using anything but a side offset key starting drift.

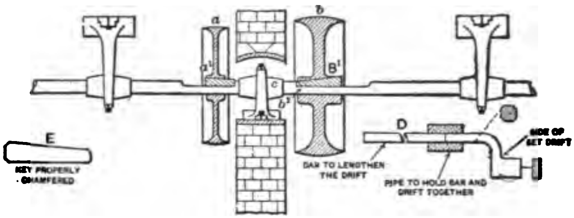


FIG. 2.

An effort was made to loosen *b* (which was farthest from the wall) by sledge-driving it toward the wall, hoping that the pulley might move off the key. The key, as was afterward found out, not having been oiled when originally driven home had rusted in place badly; though the pulley was moved by sledging, the key, secure in the pulley hub, remained there.

Ultimately one of us had to get into pulley *b*, and, removing cap *c*, hold the improvised side offset, long, starting drift *D* in place against B^1 at b^2 while the other swung the hand sledge at *a*. The entering end of the key, not having been file chamfered off, as it should have been (see *E*), our starting drift burred it up; so, after having started it, we had the pleasure of getting

into *b* to file the key end b^2 into shape so as to admit of getting it out.

The solid pulley *b* has since been replaced with a split pulley.

By the arrangement, as shown in Fig. 3, of the rim-friction clutch on the driven main shaft *B* and the driving pulley on the engine-connected driving main shaft *A*, no matter whether *B* shaft is in use or not — *i.e.*, whether the clutch be in or out of engagement — so long as *A* shaft is in motion the belt *C* is working.

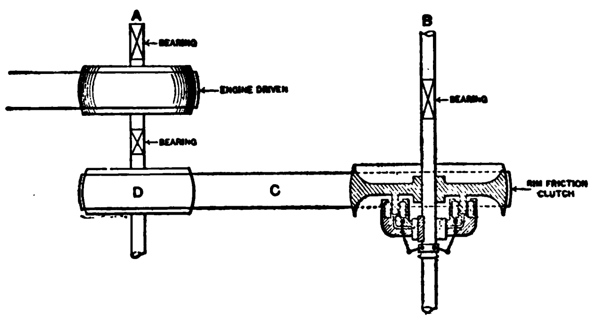


FIG. 3.

Main line belts come high, and the more they are used the sooner will they wear out. By changing the clutch from shaft *B* to *A* and the pulley *D* from *A* to *B*, belt *C* will be at rest whenever *B* is not in use. Where, however, these shafts are each in a separate room or on a different floor (the belt running through the wall or floor and ceiling, as the case may be) the clutch, despite belt wear, should be placed directly on the

driven shaft (as *B*), so as to provide a ready means for shutting off the power in cases of emergency.

Figs. 4, 5 and 6 represent a dangerous mode, much in vogue, of driving an overhead floor. An extremely slack belt connects the driving shaft *A* and the driven shaft *B;* when it is desired to impart motion to the driven shaft the belt tightener *C* is let down and belt contact is thus secured.

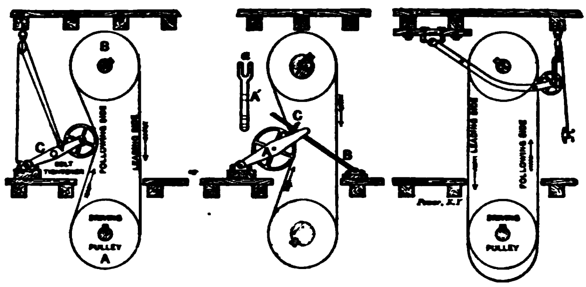


FIG. 4. FIG. 5. FIG. 6

This tightener system is called dangerous advisedly, for few are the shops employing it but that some employee has good cause to remember it. Unlike a clutch — where control of the power is positive, instantaneous and simple — the tightener cannot be handled, as in emergency cases it has to be.

In any but straight up and down drives with the driven pulley equal to or larger (diametrically) than the driver, unless the belt have special leading idlers there is more or less of a constant belt contact with its resultant liability to start the driven shaft up unexpectedly. When the tightener is completely off, the

belt, owing to heat, weight or belt fault, may at any time continue to cling and transmit power for a short space, despite this fact.

These tighteners are usually pretty heavy — in fact, much heavier than the unfamiliar imagines when on the spur of emergency he grapples them, and trouble results.

Tightener (in Fig. 5) *A* is held in place by two threaded rods *B* — as shown by slot *a* in A^1 — and regulated and tightened by ring-nuts *C* working along the threaded portion of *B*. *C* (of Fig. 4) is also a poor arrangement. Fig. 6 is the best of them all.

Apropos of clutches, great care must be exercised in tightening them up while the shafting is in motion, for if the least bit overdone the clutch may start up or, on being locked for trial (according to the clutches' structure), continue running without possibility of release until the main source of power be cut off. Nothing can exceed the danger of a clutch on a sprung shaft.

Heavily loaded shafting runs to much better advantage when center driven than when end driven, and what often constitutes an overload for an end drive is but a full load for a center drive. To illustrate, here is one case of many: The main shaft — end driven — was so overloaded that it could be alined and leveled one week and be found out one way or the other, frequently both ways, the next week. Being tired of the ceaseless tinkering that the condition under which that shaft was working necessitated, the proprietors were given the ultimatum: A heavier line of

shafting which would be sure to work, or a try of the center drive which, owing to the extreme severity of this case, might or might not work.

A center drive, being the cheapest, was decided upon. Pulley *A*, Fig. 7, which happened to be a solid, set-screw and key-held pulley, was removed from the end of the shaft. The split, tight-clamping-fit pulley *B*, Fig. 8, was put in the middle of the shaft length; the

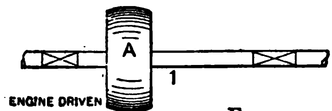


FIG. 7.

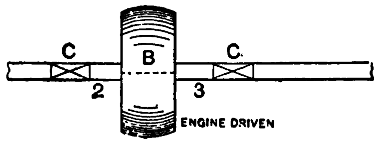


FIG. 8.

gas engine was shifted to accommodate the new drive, and hanger C^1 was put up as a reinforcement to hanger C and as a preventive of shaft springing. After these changes the shaft gave no trouble, so that, as had been hoped, the torsional strain that had formerly all been at point 1 must evidently have been divided up between points 2 and 3.

When a main shaft is belted to the engine and to a countershaft, as shown in Fig. 9, the pulley A^1 gets all the load of main and countershafts. In the arrangement shown in Fig. 10 point 1 gets *A's* load and 2 gets *B's* load and is the better arrangement.

Where a machine is situated close to one of the columns or timber uprights of the building it is very

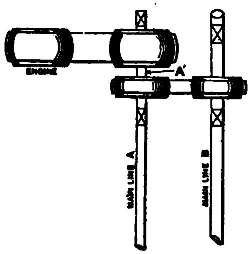


FIG. 9.

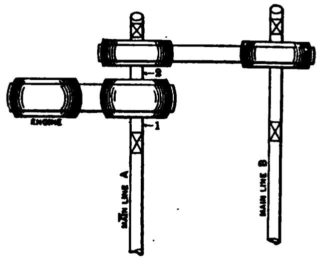


FIG. 10.

customary to carry the belt shifter device upon the column, as in Fig. 11. The sudden stoppage of a machine seldom does any damage, whereas an unex-

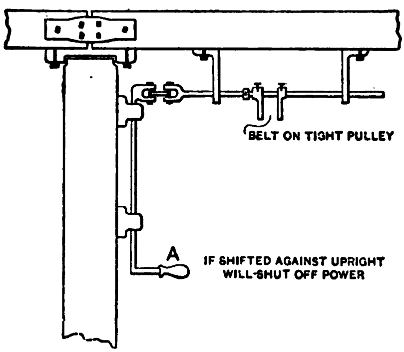


FIG. 11.

pected starting may cause irreparable damage and often even endanger the limb and life of the machine operative.

To avoid the possibility of some passing person brushing up against the shifting lever and thus starting the machine, the tight and loose pulleys of the counter-shaft should be so placed that when *A* is exposed — that is, away from the column — its accidental shifting shall stop the machine. Fig 12 makes this point clear.

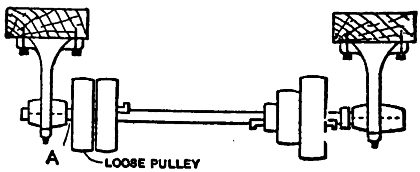


FIG. 12.

This arrangement is often used to save a collar (at *A*). The oil runs out between the loose pulley and the bearing, especially if the latter be a split bearing; the loose pulley, instead of being totally free when the belt is on the tight pulley, acts more or less, in proportion to the end play of the shaft, as a buffer between the tight pulley and the bearing; finally, the tight pulley is deprived of the support (which, when under load, it can use to good advantage) a nearer proximity to the hanger would give it.

The shafts of light-working counters should not be needlessly marred with spotting or flats for collar set-screws, nor should cup or pointed set-screws (which mar a shaft) be used. If the collar be sharply tapped with a hammer, diametrically opposite the set-screw, while it is being tightened up., all slack is taken out of the collar; and the hold is such that, without resource

to the same expedient when loosening the collar, a screwdriver will scarcely avail against a slotted set-screw.

When required to sink the head of a bolt into a timber to admit of the timbers lying snug in or against some spot, if allowable, the bolt's future turning can be guarded against by cutting the hole square to fit the bolt head. But where a washer must be used, the only positive and practical way to prevent the bolt from turning is to drive a nail (as shown) into *A* (Fig. 13) far enough for the nail head to flush *B;* now bend

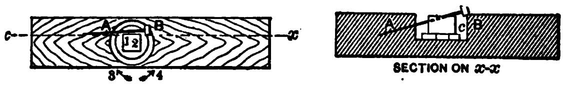


FIG. 13.

the head down behind the bolt toward *c*. It is evident that if the bolt tries to turn in the direction of 3 the nail end (wood held) will prevent it; if toward 4, the nail head will be forced against the wood and catch hold of the bolt head.

Large belts of engines, dynamos, motors, etc., when in need of taking-up are usually attended to when the plant is shut down; that is, nights, Sundays or legal holidays. At such times power is not to be had; and if the spliced part of the belt, which must be opened, shortened, scraped, re-cemented and hammered, happens to be resting against the face of one of the pulleys, is up between some beams or down in a pit, the chances of the job, if done at all, being any good are very slim.

The spliced part of a large belt should be clearly

marked in some permanent and easily recognizable way (a rivet, or where the belt is rivet-held at all its joints some odd arrangement of rivets is as good a way as any). This marking will minimize the possibility of mistake and enable the engineer to place the belt splice in the position most favorable for the belt-maker's taking-up.

In wire-lacing a belt, very often, despite all efforts and care, the edges of the belt (*A*, *B*) get out of line, as shown in Fig. 14, and make the best of jobs look poor. By securing the belt in proper position by two small pieces of wire passed through and fastened at 1, 2, 3 and 4, Fig. 15, the lacing can be more conveniently

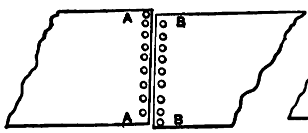


FIG. 14.

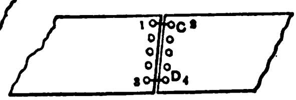


FIG. 15.

accomplished and the edge projection is avoided. When the lacing has progressed far enough to necessitate the removal of wires *c d*, the lacing already in place will keep the belt in its original position.

A wire lacing under certain conditions will run a certain length of time to a day. On expensive machinery whose time really is money it pays to renew the lacing at regular intervals so as to avoid the loss of time occasioned by a sudden giving out of the lace.

Never throw a belt on to a rim-friction or other kind of clutch while the shaft is in full motion. Belts, when being thrown on, have a knack, peculiarly their own,

of jumping off on the other side of the pulley. And should a belt jump over and off on the wrong side and get caught in the clutch mechanism, as the saying goes, "there will be something doing" and the show usually comes high. It pays to slow down.

A mule belt (transmitting in the neighborhood of or considerably over 25 horse-power) that runs amuck through the breaking down of the mule can make enough trouble in a short time to keep the most able repairing for a long while.

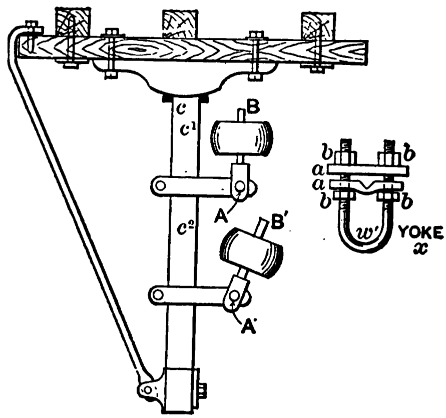


FIG. 16.

No matter what the pulley shafts holding arrangement and adjusting contrivance may be, all of the strain due to belt weight, tension, and the power transmitted falls mainly at points A, A^1, Fig. 16; and it is here that, sooner or later, a pin, set-screw or bolt gives way and the belt either gets badly torn up, rips

something out of place, or a fold of it sweeping to the floor slams things around generally until the power is shut off.

The remedy is obvious: Reinforce *A*, *A′* by securing *B*, *B′* to the supporting shaft *c* at c^1, c^2. The yoke *x* is a reliable and practical means to this end. Straps *a* held by the nuts *b* hold the yoke securely on the supporting shaft *c*, while the pulley-shaft ends *B*, *B′* are held in the *U* of the yoke at *w′* at any desired distance from *c* by means of the adjustment provided by the nuts *b*.

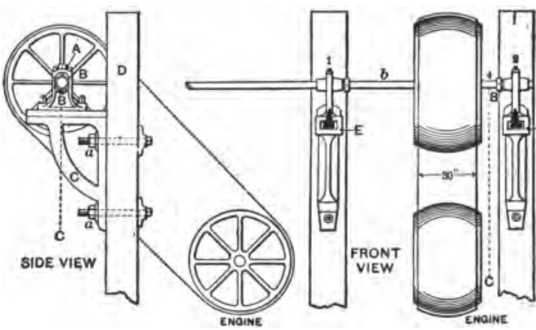
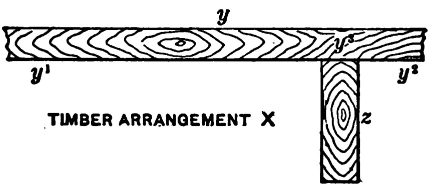


FIG. 17.

The end of a hanger bearing was badly worn (Fig. 17). The cap could be lifted out by removing bridge *A*, but the shaft interfered with the lifting of the bottom out, owing to its being held in the hanger slides. It had to be removed and we were called upon to put it into shape by re-babbitting.

Being a newspaper plant, money was no object; the time limit, however, was three hours, or hands off.

Opening the 30-inch engine belt and removing the interfering shaft length was out of the question in so short a time. So the job was done as follows: The shaft was braced against down sag and engine pull along the line *B C* by a piece of timber at *A*, and against pull on *B D* by timber arrangement *X*; timber *y*'s points y^1



and y^2 resting against the uprights at 1 and 2, timber *z* wedged in between *y* at y^3 and the shaft at 4, thus acting as the stay along line *B D*. The nuts and washers *a, a* were removed; the bolts driven back out of the bracket; the end of a rope was thrown over the shaft at *b*, passed through the pulley and tied to the bracket and hanger which, as one piece, were then slid endways off the shaft and lowered to the floor. The bearing was cleaned, re-babbitted and scraped, everything put back, stays removed and the shaft running on time with a half-hour to the good.

When desirable to keep a shaft from turning while chipping and filing flats, spotting in set screws or moving pulleys on it, it can be done by inserting a *narrow* strip of cardboard, soft wood or several thicknesses of paper between the bearing cap and the top of the shaft and then tightening the cap down.

The packing, 1-16 to 3-16 inch thick and about as

long as the bearing, must be narrow; otherwise, as may be deduced from Fig. 18 (which shows the right way), by the use of a wide strip in the cap the shaft is turned into a wedge, endangering the safety of the cap when forced down. At point 3 packing does no harm, but at 1 and 2 there is just enough space to allow the shaft diameter to fit exactly, with no room to spare, into the cap bore diameter.

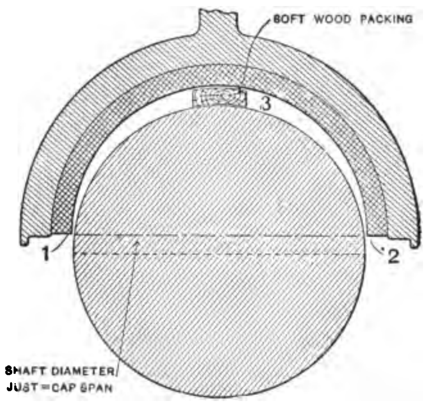


FIG. 18.

As a very little clamping will do a good deal of holding the clamping need not be overdone. A shaft can also be held from turning, or turned as may be desired, by holding it with a screw (monkey) wrench at any flat or keyway, as shown in sectional view, Fig. 19.

When a shaft breaks it is either owing to torsional strain caused by overload, springing through lack of

hanger support at the proper interval of shaft length, the strain of imperfect alinement or level, or a flaw.

An immediate temporary repair may be effected by taking some split pulley that can best be spared from another part of the shaft and clamping it over the broken part of the shaft, thus converting it, as it were, into a compression coupling. The longer the pulley hub the better the hold; spotting the set-screws — that is, chipping out about ⅛-inch holes for their accommodation into the shaft — is also a great help.

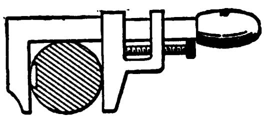


FIG. 19.

If when the shaft breaks it has not been sprung by the sudden dropping of itself and the pulleys that were on it, a permanent repair can be effected, after correcting the cause of the break, by the use of a regular keyless compression coupling.

If it has been sprung, a new length comes cheapest in the wind-up; and if overload was the original cause of the trouble, only a heavier shaft or a considerable lightening of the load will prevent a repetition.

In Fig. 20 *A* shows how to drive to make belt weight count in securing extra contact. In *B* this weight causes a loss of contact. Bearing in mind that *B* is not only a loss from the normal contact but also a loss of the extra contact that *A* gives, it will readily be seen how important a power-saving factor the right sort of

a drive is — especially on high-speed small-pulley machines, such as dynamos, motors, fans, blowers, etc.

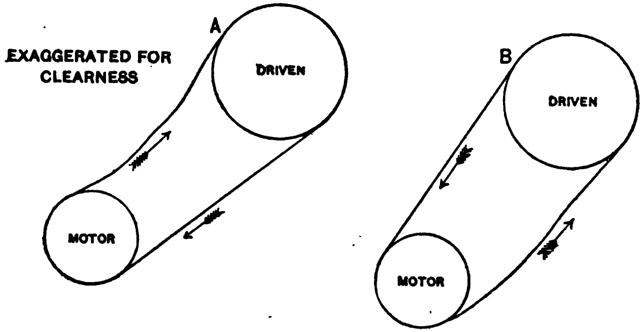


FIG. 20.

A good many electrical concerns mount some of their styles of dynamos and motors (especially the light duty, small size) upon two *V*-shaped rails, Fig. 21 (the bottom of the motor or dynamo base being V-grooved for the purpose). The machine's weight and

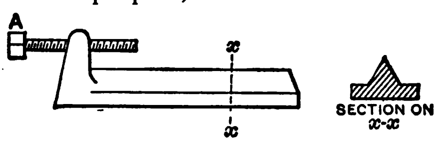


FIG. 21.

the screws *A* are counted on to keep it in place. If the machine be properly mounted on these rails, as regards screws *A* in relation to its drive, the screws reinforce the machine's weight in holding it down and also permit a surer adjustment through this steady holding of the machine.

Fig. 22 shows the machine properly mounted. The belt tension and pull tend to draw *B* corner of the machine toward the shaft *C;* and screw B^1 is there to resist this pull. Owing to this resistance and the pull along line *D, E* tends to lift and slew around in E^1 direction; screw E^2 is, however, in a position to overcome both these tendencies. If the screws are both in front, there is nothing but the machine's weight to keep the back of it from tilting up. The absurdity of placing the screws at *F* and *G,* though even this is thoughtlessly done, needs no demonstration.

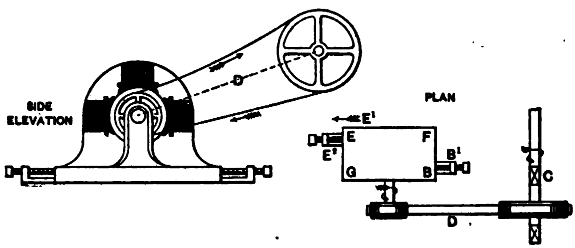


FIG. 22.

When putting a new belt on a motor or dynamo, both the driver and the driven are often needlessly strained by the use of belt-clamps, in the attempt to take as much stretch out of the belt as possible. On being loosely endlessed it soon requires taking up; and if only laced, when the time for endlessing comes the belt is botched by the splicing in of the piece which, owing to the insufficiency of the original belt length, must now be added to supply enough belt to go around, plus the splice.

The proper mode of procedure is: Place the motor on its rails or slides 5 inches away from its nearest possible approach to the driven shaft or machine and wire-lace it (wire-lacing is a very close second to an endless belt). Let it run for a few days, moving the motor back from the driven shaft as the belt stretches. When all reasonable stretch is out, move the motor back as close to the driven shaft as possible.

The 5 inches forward motion will give 10 inches of belting, which will be amply sufficient for a good splice; and, further, the machine will be in position to allow of tightening the belt up, by simply forcing the motor back, for probably the belt's lifetime.

II

SHAFTING HINTS[1]

The bolts, set-screws, pulleys, bearings, shafting and clutches of a plant, although among the foremost factors in its efficiency, are very often neglected until they reach the stage where their condition absolutely compels attention.

Very often this lack of proper attention is due to surrounding difficulties of an almost insurmountable and most discouraging nature. At other times it is due to a lack of proper appreciation of the damage resultant from seemingly insignificant neglects. How to overcome some of these difficulties is the object of this chapter.

Fig. 23 shows a case of a turning bolt. The head is inaccessible and the bolt's turning with the nut, owing to burrs or rust, prevents either the tightening or the loosening of the nut. One to three fair-sized nails driven through the timber as at *C*, hard up against, or, better still, forced into a tangent with the bolt, will often suffice to hold it while the nut is being turned. In iron girders, beams, etc., the nail method being impossible, a slot *E* can easily be cut with a hack-saw through the lower end of both the nut and bolt, so

[1] Contributed to Power by Chas. Herrman.

that the bolt may be held by a screwdriver while the nut is turned with a wrench.

Where an extra strong screwdriver must be used, the use of two blades at the same time in the hack-saw frame will give a slot of the requisite width. Where the bolt's end projects beyond the nut and it is desired to tighten the nut, a Stillson wrench is often, though inadvisedly, called into service. This tends to spoil the lower threads of the bolt and thus prevents any future loosening, except by the cutting off of the projecting end.

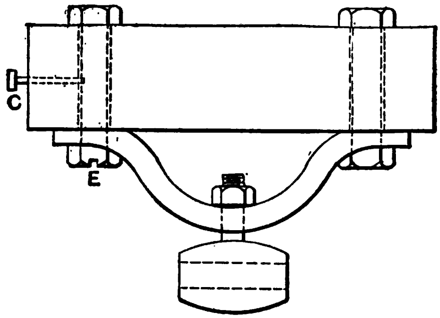


FIG. 23.

As the alinement and level of shafting depend on the power of their hold, bolts, lag-bolts and set-screws should, when they are tightened, be so in fact and not in fancy.

The proper way to use a wrench, especially a screw wrench, so as to avail yourself of every ounce of power, not of your biceps only but of your whole body, is as follows: Place your shoulders on a level with the object to be tightened, secure the wrench jaws well upon it,

grasp the jaws with the left hand and the wrench handle with the right, holding both arms straight and tense; swing the upper part of the body to the right from the hip, backing the force of your swing up with the full force of your legs, steadying yourself the while with your left-hand grip on the wrench jaws, which are the center of your swing. Several such half turns, at the wind-up, will cause an extremely hard jam with comparative ease.

In tightening up a split-pulley, the expedient of hammering the bolts tight, by means of an open-ended bolt-wrench and a small sledge, is often resorted to. If the head of the bolt be lightly tapped while the nut is being tightened, even a light hammering, except in the extremest cases, becomes unnecessary.

Split-pulleys are invariably better held in place by a good clamping fit than by set-screws. It must also be borne in mind that, for good holding, set-screws must be spotted into the shaft, and this defaces and often materially weakens the shaft. Split-pulleys, like solid ones, are sometimes subject to stoppage, owing to excessive strain. Set-screws, at such times, cut a shaft up pretty badly; whereas, if clamped, only a few slight scratches would result.

Where packing with paper, cardboard, emery cloth or tin becomes necessary to secure a good clamping fit, care should be taken to put an equal thickness of packing into both halves of the pulley; otherwise it will wabble and jump when running.

Emery cloth, on account of its grittiness, is preferable for packing where the duty done by the pulley

is light. When the duty done is extra heavy, emery cloth, despite its grittiness, will not do; tin or sheet iron, owing to body, must be used.

The following is the most practical way of packing a split-pulley to a good clamping fit, assuming that emery cloth is to be used:

The thickness of the emery cloth to be used, and whether to use one or more folds, can readily be ascertained by calipering the shaft diameter and pulley bore, or by trial-clamping the pulley by hand. In both of these instances, however, due allowance must be made for the compressiveness of the packing used. If the packing be too thin, the pulley will not clamp strongly enough; if too thick, the chances of breaking the lugs when drawing the bolts up are to be apprehended.

Having determined the proper thickness of emery cloth to be used, place the pulley on the shaft, as shown in Fig. 24. Into the lower half *C*, in space *A*, which is out of contact with the shaft, place a sheet of emery with the emery side toward the hub and the smooth side toward the shaft. The width of the emery should be a little less than half of the shaft's circumference, and it should be long enough to project about one-half of an inch to an inch on each side of the hub.

Now turn the pulley on the shaft so that the position of the halves shall become reversed (Fig. 25), *C* on top, *B* on bottom. See that the emery cloth remains in its proper position in half-hub, the smooth side being toward the shaft; the projecting length beyond the pulley hub will help you to do this.

Into half-hub *B* (space *D*) insert a similar sized piece of emery cloth, smooth side toward the hub and the emery side toward the shaft. Draw up on your bolts to clamp the pulley into position. Be sure, however, that no emery cloth gets in between the half-hubs or lugs at points 1 and 2, Fig. 25, as this would prevent their coming properly together; the width of the emery being less than half of the shaft's circumference will be a help to this end.

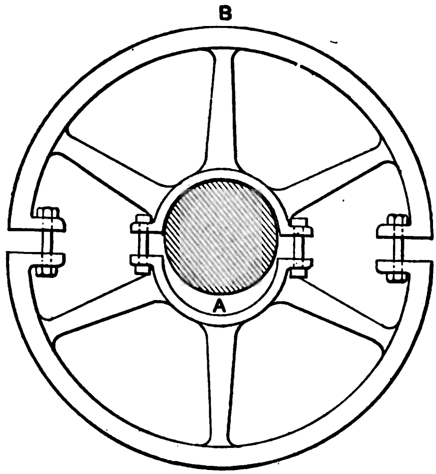


FIG. 24.

It often happens, owing to downright neglect or unwitting neglect, through the oil hole or oiler being blocked up, that a loose pulley, running unlubricated, cuts, heats, and finally, through heat expansion, seizes. It then becomes necessary to take the countershaft

down, force the loose pulley off and file and polish the shaft up before it can be put back into place.

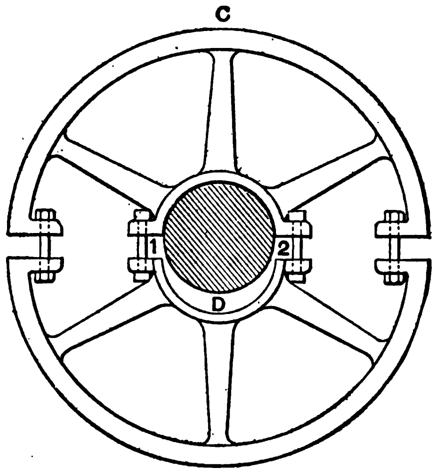


FIG. 25.

The following method avoids the taking down and putting back, provides an easy means for loosening up the pulley that has seized, and improvises, as it were, a lathe for filing and polishing the shaft.

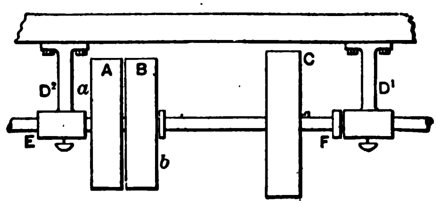


FIG. 26.

In Fig. 26, *A* is the loose pulley that has seized.

Throw off both the belt that leads from the main shaft to pulleys *A*, *B* and the belt that leads to the driven machine from the driving pulley *C*. Tie, or get somebody to hold, an iron bar in pulley *A* at side *a*, as shown in Fig. 27, over an arm of the pulley, under the shaft,

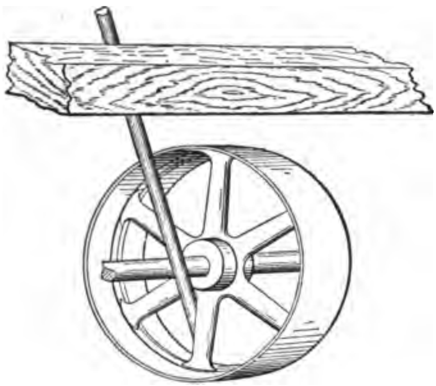


FIG. 27.

and resting against the timber, ceiling, wall or floor, in such a way as to prevent the pulley from turning in one direction, as shown in Fig. 27. Now, with another bar, of a sufficient length to give you a good leverage, take the grip under a pulley arm and over the shaft in the tight pulley *B* at *b*, which will enable you to work against the resistance of the bar in the loose pulley *A*.

With enough leverage, this kind of persuasion will loosen the worst of cases. Take the bars out and move *B* sufficiently to the right to allow *A* to take *B*'s former position. Secure *B* by means of its set-screws in its new position and, by means of a piece of cord, fasten

an arm of *A* to one of *B*'s. It is evident that by throwing the main-shaft belt on to *A* it will, through *A*'s cord connection with *B*, which is screwed to the shaft, cause the shaft to revolve, thus enabling you to file up and polish that portion of it formerly occupied by *A*. To prevent the countershaft from side-slipping out of hanger-bearing D^1, get somebody to hold something against hanger-bearing D^2 at *E;* or fasten a piece of wire or cord on the countershaft at *F* and the hanger D^1, so as to prevent side-slipping while not interfering with revolution.

Filing, polishing, a cleaning out of the oil hole or oiler, and the taking of proper precaution against future failure of lubrication will put everything into first-class order. When the loose pulley is, as it is best for it to be, farthest away from the bearing, held in its place by the tight pulley and a collar, not only is the tight pulley better adapted for carrying its load, owing to additional support resultant from its proximity to the bearing, but such matters of small repair as come up are much simplified.

Fig. 28 in some degree, aside from the cutting up and

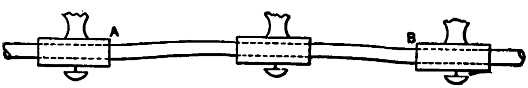


FIG. 28.

heating of the bearings, illustrates the breaking strain, in addition to the usual torsional strain, which becomes enhanced in direct proportion with the increase of breaking strain, to which an out-of-line or out-of-level

shaft is subject. The bends are exaggerated for illustration.

In this instance, the fact of one hanger-bearing being out of line or level subjects the shaft to a severe breaking strain. The shaft being both out of line and level does not, if both at the same point, aggravate matters, as might at first be supposed.

It is true that the full torsional strength of a shaft is only equal to the weakest portion of it, so that three weak spots more or less can, theoretically, make no difference one way or the other. But, practically, there is the undue strain and wear of the bearings at these points, and if a pulley transmitting any considerable amount of power is situated anywhere along the length *A B* it is sure to be unpleasantly in evidence at all times.

Only an eighth or a quarter out, but oh, what shaft-breaking stories that fraction could tell!

The following is a simple method for testing the alinement and level of a line of shafting that is already up.

As in Fig. 29, stretch a line *C* so that it is exactly

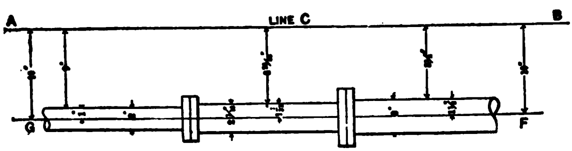


FIG. 29.

opposite the shafting. Set it equidistant from the shaft end centers *G* and *F* and free from all contact along its entire length except at its retaining ends *A* and

B. Now, it is self-evident, as line *C* is straight and set equidistant from the shaft end centers *G* and *F*, that if you set the entire center line of the shafting at the same distance from line *C*, as *G* and *F*, you are bound to get your shafting into perfect alinement.

In leveling a line of shafting that is already up, you can, by the use of a level and perseverance, get it right.

Placing the level at *A*, you are just as likely to raise the first hanger as to lower the middle one. Look before you jump, even if compelled to climb to the top of the fence to do so. When you find a length of shafting out of level, try the two adjacent lengths before acting, and your action will be the more intelligent for it.

On exceptionally long lines of shafting the following method, in which the level and a line constitute a check upon and a guide for each other, can be used to great advantage. Stretch a line so that it is exactly above, or, if more convenient, below the shafting to be leveled. With the level find a length of shafting that is level and adjust your line exactly parallel with this length. Your line now, free of contact except at its retaining ends, and level owing to its parallelism to the level shaft length, constitutes a safe *hight level* guide while the level itself can serve to verify the accuracy of the finished job.

In lining, whether for level or alinement, unless the shafting line consists of the same diameter of shafting throughout its entire length, though of necessity measuring from the shaft circumference to the line, always

base your calculations on the shaft centers. The figures in Fig. 29 will make this point clear.

The manner of securing the ends of the line under different circumstances must be left to individual ingenuity. Only be sure that the line is so placed that the shafting adjustment shall not affect its original position with reference to the end shaft centers.

Coupling clutches, *i.e.*, those joining two lengths of shafting into one at option, will fail, utterly or partially, if the respective shafts which bear them are out of line or level with each other. Such a condition should not be tolerated on account of the danger entailed by the inability to shut off the power in cases of emergency.

As a general rule, it is most advisable to set a clutch to take as hard a grip as it can without interfering with its releasing power. Where a clutch grips weakly, it is subject to undue wear owing to slippage, whereas a strongly regulated clutch absolutely prevents slippage wear.

III

SHAFTING HINTS[1]

ENGINEERS, machinists and general mechanics are often called upon to turn their hands to a shafting job. We recognize that all of the following cannot prove new or even suggestive to most of our readers; still, some of it for all, and, mayhap, all for some, may not come amiss.

We all know that to have belting run rightly on pulleys located upon parallel lines of shafting the shafting must be in absolutely correct parallel. The slightest deviation, even to a 1-16 inch, often imparts a marring effect, through poorly running belts, to an otherwise faultless job.

Fig. 30 shows how to line a countershaft as regards

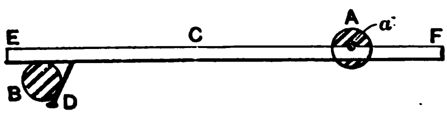


FIG. 30.

parallelism with the driving shaft when the countershaft's end-centers are availably situated for thus measuring. *A* is the countershaft, *B* the main shaft, *C* is a stick of proper length about 1½ inches in thickness

[1] Contributed to Power by Chas. Herrman.

and width, *D* a heavy nail — about 20-penny will do — driven into *C* far enough from its end *E* to allow of *C's* resting squarely upon the top of the shaft *B*.

Rest the measuring rod upon the main shaft, keeping the nail in touch with the shaft, so that when the *F* end is in contact with the end of the countershaft the stick shall be at right angles to the main shaft, and then mark the exact location *a* of the countershaft's end-center on the stick. Do the same at the other end of the countershaft. If both marks come at the same spot, your counter is parallel; if not, space between these two marks will show you how much and which way the counter is out.

It may only be necessary to shift one end in or out a little; and then, again, it may be that to get into line you will have to throw one end all the way in one direction and the other all or some in the opposite direction. But, whichever it be, do not rest content until you have verified the correctness of your adjustment by a re-measurement.

The nail should be well driven into *C*, so that its position will not readily change, and it should, preferably, be slant driven (as shown in Fig. 30), as it thus helps to keep the stick down in contact with the shaft.

Where an end-center is not available or where there is no clear space on the main shaft, opposite a center, the method shown in Fig. 31 can generally be used.

Rest *C* on top of both shafts and at right angles to the driving shaft *B*. With *D* pressed against *B*, place a square on stick *C*, as shown (stock in full contact with the top of the rod, and the tongue running down the

side of it). Slide along *C* toward *A* until the side of the tongue touches the shaft the other side of *A*. Now mark a line on the stick down tongue. Do the same at the other end of your countershaft and the two resultant marks will be your parallel adjustment guides.

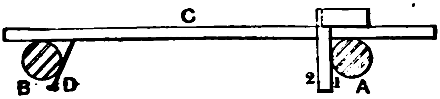


FIG. 31.

It often happens that a counter, or even line shaft, is end driven from the extreme end of the main or jack driving shaft with its other end running beyond the reach of the driving shaft, as shown in Fig. 32.

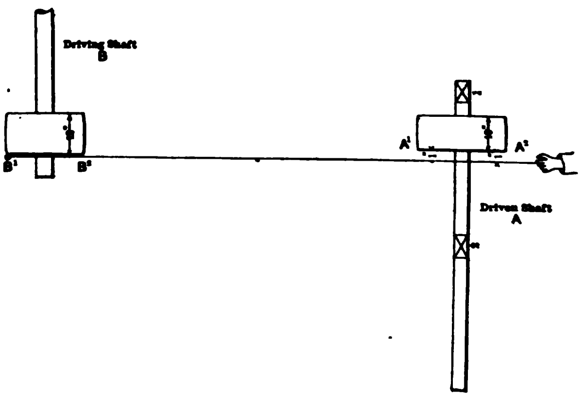


FIG. 32.

It is evident that neither method 1 nor 2 can here be applied to solve the alinement problem. If the

driving pulley *B* and the driven pulley *A* are both in place, the following method can be used to advantage.

Fasten, or let somebody hold, one end of a line against pulley *B*'s rim at B^1; carry the line over to *A* at A^2; now sweep the loose A^2 end of the line toward pulley *A* until the line just touches pulley *B*'s rim at B^2. When the line so touches — and it must just barely touch or the measurement is worthless — A^1 and A^2 of pulley *A* must be just touched by or (if *B* and *A* are not of a like face width, as in Fig. 32) equidistant from the line.

A single, two-hanger-supported length of shafting thus lined is bound to be in parallel; but where the so adjusted shaft line consists of two or more coupling-joined lengths supported by more than two hangers, only pulley *A*'s supporting portion of the shaft between its immediate supporting hangers 1 and 2 is sure to be lined; the rest may be more or less out.

To make a perfect job, fix a string in parallel with shaft length 1 and 2, stretching along the entire length of the adjusted shaft, and aline the rest of the shaft length to it.

When there are no pulleys in place to go by, or when, as occasionally happens, the wabbly motion of pulley *B* (when running) indicates that, having been inaccurately bored or bushed, or being located on a sprung shaft length, its rim line is not at right angles to the shaft line, the method shown in Fig. 33 can be resorted to.

Instead of the nail used in methods 1 and 2, use a board about 8 to 12 inches long and of a width equal to considerably more than half of shaft *B*'s diameter.

By nailing this board *x* to the measuring rod *c* at any suitable angle, you will be enabled to reach from the end *a* well into the shaft *B*, as at *b*, and from *b'* well into *A*, as *a'*. By keeping the board *x* along its entire length in full contact with the shaft *B* at both 1 and 2, the angular position of rod *C* is bound to be the same in both instances, and you will thus (by the use of a square, as in Fig. 31) be enabled to aline *A* parallel with *B*.

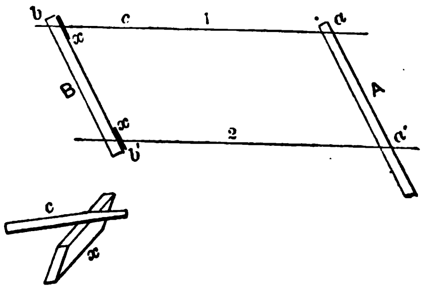


FIG. 33.

In all instances of parallel adjustment here cited it is assumed that both the alined and the alined-to shafts have been, as to secure accuracy of result they must be, properly leveled before starting to aline.

The above methods apply to cases where the shafting is already in place. Where, however, shafting is being newly installed before the work can be proceeded with, it is necessary, after determining on the location for the shafting, to get a line on the ceiling in parallel with the driving shaft to which to work to. Mark that point *A* which you intend to be the center line for the proposed shafting upon the ceiling (Fig. 34).

Rest your measuring rod upon the driving shaft and at right angles to it, with the nail against it. Hold your square with the stock below and the tongue against the side of the measuring stick, so that its tongue extremity touches the ceiling mark *A*, and then mark a line on the rod along the tongue side *A*. Move your rod along the driving shaft to the point where the other end of the proposed shafting line is to be, and, squaring your stick to the driving shaft with the tongue side *A* on the marked line of the stick, mark your section point on the ceiling. Draw a line or stretch a string between these points, and you have a true parallel to work to.

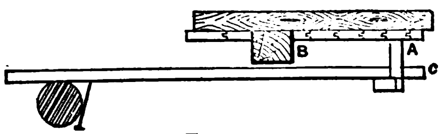


FIG. 34.

Owing to the supporting timber *B*'s interference, a square had to be used; but where the ceiling is clear the rod can be cut to proper length or the nail be so located as to allow of using the stick extremity *C* for a marking point.

When a pulley is handily situated on the driving shaft, the method shown in Fig. 35 can be used to advantage.

Let somebody hold one end of a line at 1, and when you have got its other end so located on the ceiling that the line just touches the pulley rim at 2, mark that ceiling point (we will call it 3). In the same way get your marks 4 and 5, each farther back than the other

and, for the better assurance of accuracy, as to just touching at 2, remove and readjust the line separately each time. If now a straight line from 3 to 5 cuts 4, your line 3, 4, 5 is at right angles to the driving shaft and a line at right angles to this will be parallel to the shaft.

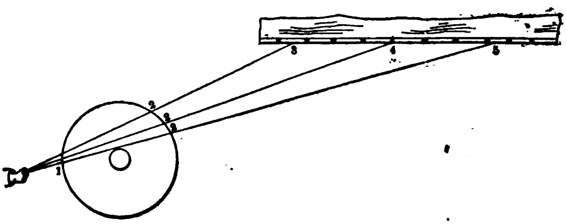


FIG. 35.

The plumb-bob method is so familiar and, where not familiar, so easily thought out in its various applications, that we deem it useless to touch upon it.

The stringers or supporting timbers of drop hangers should be equal in thickness to about one-fifth of the hanger drop.

Where the stringers run with the hangers and crosswise of the shaft, both feet of a hanger base are bolted to the same stringer, and this should be from $1\frac{1}{4}$ to $1\frac{1}{2}$ times the width of the widest portion of the hanger base. As the hanger is securely bolted to its stringer, this extra width is in effect an enlargement of the hanger base, and thus enables it the better to assist the shaft's end motion.

Where the stringers run with the shaft and crosswise of the hangers, the two feet of the hanger base are each fastened to a separate timber, and these should

be equal in width to the length of one hanger foot, plus twice the amount of adjustment (if there be any) the hanger's supporting bolt slots will allow it. In reckoning hanger adjustment, be sure to figure in the bolt's diameter and to bear in mind that to get the utmost adjustment for the countershaft the bolts should originally be centered in the slot; thus a 1⅜ × 1½-inch slot, as it calls for a ¾-inch bolt, leaves a ¾-inch play,

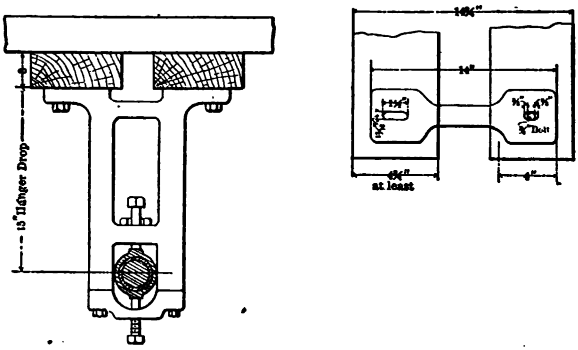


FIG. 36.

and this play, with the bolt in the center of the slot, allows of ⅜-inch adjustment either way. Without this extra width addition any lateral adjustment of the hanger would result in leaving a part of the hanger's feet without stringer support. Such jobs look poorly, and often run still more poorly. Fig. 36, in its two views, will make the above points clear.

In the stringing of countershafts whose hangers have no adjustment it often happens, despite all care

in the laying out, that they come $\frac{1}{8}$ to $\frac{1}{4}$ inch out of parallel. A very common and likewise very dangerous practice at such times is to substitute a smaller diameter supporting bolt instead of the larger size for which the hanger foot is cored or drilled, and to make use of the play so gained for adjustment.

That shafting so carried does not come down oftener than it does is due solely to the foresight of the hanger manufacturers. They, in figuring the supporting bolt's diameter as against the strain and load to be sustained, are careful to provide an ample safety margin for overload, thus enabling the bolt substituted to just barely come within the safety limit under easy working conditions.

The largest-sized bolt that a hanger will easily admit should invariably be used, and for alinement purposes either of the following slower but safer methods should be used.

Rebore the hanger-supporting bolt holes in the stringers to a larger size, and use the play so gained for adjustment. It is not advisable, however, to rebore these holes any larger than to one and three-quarter times the diameter of the bolt to be used; and the diameter of the washers to be used on top of the stringers should be diametrically equal to at least twice the size of the rebored holes. That the washers used, under such conditions, must be of a good proportionate thickness goes without saying.

When the reboring method cannot be used — as when the hangers are carried by lag screws, lag-bolts, bolts screwed directly into supporting iron girders,

etc. — it is evident that hanger adjustment can be secured by packing down one foot of the hanger base, as shown in Fig. 37.

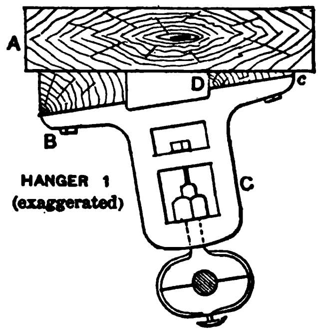


FIG. 37.

The piece of packing (necessarily wedge-shaped) between the hanger foot *B* and the stringer *A* tilts the bottom of the hanger forward. The size of the wedge regulates the amount of adjustment. Wedge-shaped space *D*, at foot *C*, should also be packed out so as to avoid throwing undue strain upon *C*'s extremity *c*. If now, the foot *c* of the countershaft's other supporting hanger (No. 2) be similarly and equally packed, as *B* of No. 1 hanger, the shaft will have been thrown forward at one end and back at the other, and thus into line. The equal division of the adjusting wedge packing between the opposite feet of the two hangers enables a limited packing to do considerable adjusting without any undue marring effect; and, further, insures the shaft's remaining level, which evi-

dently would not be the case if only one hanger were packed down.

After so adjusting, be sure to get your hangers squarely crosswise of the shaft as readjusted, so that the hanger bearings will lie in a true line with the shaft and not bind it. At all times be sure to have your hangers hang or stand plumb up and down; as, if the bearings are not so pivoted as to be horizontally self-adjusting, excessive friction will be the lot of one end of the bearing with not even contact for the rest of it. The bearing being self-adjusting all ways, square crossing of the shaft line by the hanger line and plumb still remain eminently desirable for appearance's sake.

Before a countershaft can be put up on a ceiling whose supporting timbers are boarded over, or in a modern fireproof structure whose girders and beams are so bricked and plastered in as not to show, it is necessary to positively locate those of them which are to carry the stringers.

It is in the earnest endeavor to properly locate these that the unaccustomed hand turns a wood ceiling into a sieve and a brick one into a wreck. To avoid kitchen and house razing effects, try the following recipe:

We will assume that line *A B*, Fig. 38, laid out by one of the methods previously described, is the center line of the proposed countershaft. The hanger's base length, lateral adjustment and individual foot length call for stringers $4\frac{3}{4}$ inches wide, placed $5\frac{1}{4}$ inches apart or $14\frac{3}{4}$ inches outside (as per sketch). The floor position of the machine to be driven, or the driving point of the main shaft, is so located with reference to the counter-

shaft that one of the supporting hangers must go at or very near *C*, and the countershaft's length brings the other hanger at or very near *D*.

Now between points *C D* and with due reference to the center line *A B*, lay out the position which your stringers are to occupy. It is self-evident that by confining your beam prospecting to the stringer spaces *E* and *F*, ultimately, when the countershaft is in place, all the cut-up portions of the ceiling will be hidden from view.

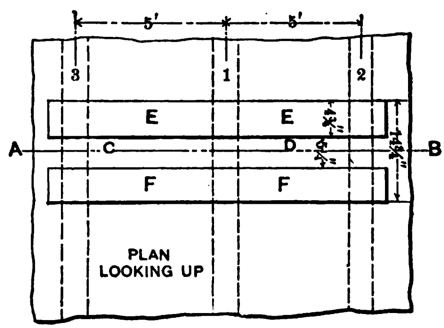


FIG. 38.

Generally the necessary supporting beams will not all be found within the shaft's length distance *C D;* in such cases continue your cutting in the same parallel line to *A B*, as at *E* or *F*, going from *C D* outwardly until you strike the sought-for beams. Having located beams, say 1 and 2, we find by measurement that they are 5 feet apart, and, as beams are generally uniformly spaced, we may start 4 feet 6 inches (go 4 feet 6 inches and not 5 feet, to make sure not to skip beam 3 and

thus make a cut that will not be covered by the stringers) from 1 to cut outwardly for the location of beam 3.

Where the building's beams run parallel to the shaft, Fig. 39, mark the counter's-center line *A B*, and then mark the spaces — as determined by the countershaft length, floor position of the driven machine or the driving point on the main shaft — to be occupied by the stringers *C D*, and, starting from the center line *A B*, cut outwardly each way to the desired beams 1 and 2.

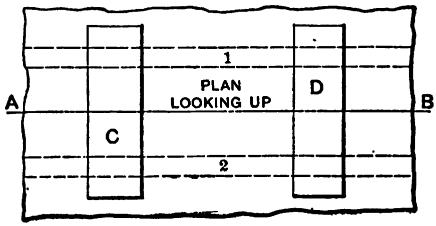


FIG. 39.

Where the center line as laid out (before the position of the ceiling beams was known) brings it close to or directly under a supporting beam, it is generally advisable where possible to step the counter back or forward to a central position between the beams.

Where shafting is already in place in a building, no matter on what floor, valuable measurements as to beam location can thus be had from the plainly in sight and the reasonably deducible. Lacking in-place-shafting to go by, the walls, columns and main girders always clearly indicate the crosswise or parallel run of the ceiling beams to the proposed shafting line.

In the usual method of locating the timbers of a boarded-over ceiling, a brace and bit, or a nail, can be used for the purpose. If shy of an awl, and in preference the other two ways, force or drive a chisel (cold chisel or wood) in between a tongue and groove of the ceiling boards in stringer space (Fig. 38) *E* or *F*, and thus spring the boards sufficiently apart to insert a compass saw. With the extremity of a 12-inch saw a very little cutting (along the tongue and groove, as this shows least) will enable you to locate a beam, since they generally run 8, 12, 16, 20, 24 and 30 inches apart.

Always, on locating your beam, run the point of your compass saw down the whole of the timber's width, so that any nailed-on pieces will not lead you into a false estimate of the beam's thickness.

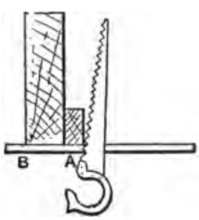


FIG. 40.

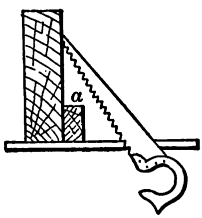


FIG. 41.

Figs. 40 and 41 make this point and its object clear. The saw, in Fig. 40, being stopped by *A*, naturally leads to the inference that *A B* is the timber's thickness. By running down the timber, as in Fig. 41, the saw's point sticking at *a* acts as a sure detector. This precaution should be taken on both sides (*B* and *A*) of the timber, and then, when the lags are screwed in,

they can be sent home safe and true in the center of the timber.

It often happens that in boring for the lag screws the bit strikes a nail and further progress at that point seems out of the question. When so situated, take your bit out, and running the lag screw up as far as it will go, by sheer force swing it three or four turns up further than the point where your bit struck. Removing the lag and replacing the bit, it will be found that the nail has been forced aside and the way is now clear.

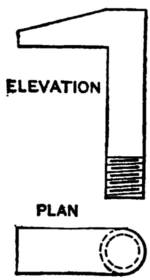


FIG. 42.

Hook bolts (Fig. 42) or — as our across-the-sea cousins call them — "elbow bolts," despite all assertions to the contrary, are an easy, safe and economical stringer fastener or suspending device.

Figs. 43 and 44 illustrate two very common abuses of the hook bolt. In the one (Fig. 43), instead of the bolt proper lying snug up against the beam flange with the whole of its hook resting squarely upon the beam's flange, its supporting countershaft is turned into a menace to limb and life by this "chance it" kind of

erection. In the other (Fig. 44), though the bolts do lie snug against the flange, the hook being out of sight and no means being provided for telling whether

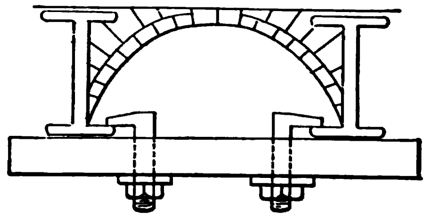


FIG. 43.

the hook lies, as it should, at right angles to the web of the beam, even if properly placed at installation, timber shrinkage, vibration or a slight turn of the bolt

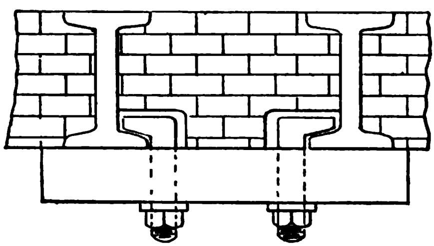


FIG. 44.

when tightening the nut, all constitute dangerous factors tending to loosen or entirely loosen the hook's grip upon the beam flange.

Fig. 43 suggests its own remedy. As to Fig. 44, a screwdriver slot (made by a hacksaw) at the nut end of the hook bolt and running in the same direction as

the hook, Fig. 45, will at all times serve to indicate the hook's position and, allowing as it does of a combined use of screwdriver and wrench, it can be used to prevent the bolt's turning when being tightened.

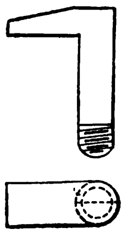


FIG. 45.

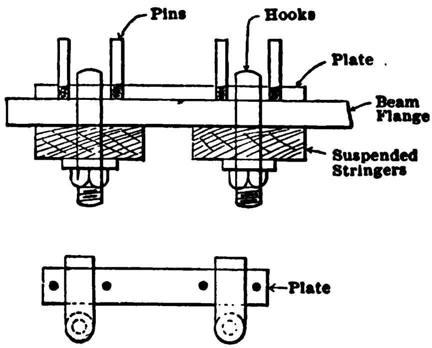


FIG. 46.

Where two or more hook bolts are placed close together on the same beam flange, a plate, preferably wrought iron with properly spaced confining pins for the hooks, may be placed between the beam flange and the hooks as in Fig. 46. Its benefits are obvious and so likewise is the use of a small, square, wrought-iron plate with a bolt hole through its center instead of hook bolts.

The various styles of beam clamps carried by the hardware and supply trade all have their good points, and though the *C* of their cost may seem to loom large, it is not a whit more emphatic, taken all in all, than the *W* of their worth.

IV

TRUING UP LINE SHAFTING

It is assumed, for the purposes of this description, that the modern style of shafting, increasing in diameter by the $\frac{1}{2}$ inch, is used, and that all pulleys and belts are in place. We will take a line composed of sizes ranging between $3\frac{15}{16}$ and $2\frac{7}{16}$ inches. This gives us four sizes, $3\frac{15}{16}$, $3\frac{7}{16}$, $2\frac{15}{16}$ and $2\frac{7}{16}$ inches in the line.

We will first consider the plumb-bob. The accompanying sketch, Fig. 47, illustrates a good one.

The ball is $1\frac{1}{2}$ inches diameter, and the large end of the tapered stem $\frac{1}{2}$ inch in diameter, turned parallel for a short distance at the lower end. The two thin sheet-steel disks, 1 and 2 inches in diameter, are drilled to fit snugly when pushed on to the $\frac{1}{2}$-inch part of the stem, and stay there until pulled off. These disks are turned true. This arrangement of plumb-bob and disks enables us to deal with five sizes on one line, and there are not many lines that contain more.

Now having our plumb-bob ready, we will stretch the line. The stretchers should be set horizontally by nailing a strip of wood, say $1 \times 1\frac{1}{2} \times 12$ inches, with a piece at each end to form a space between it and the wall, or place of location in line with the edge of the shaft, as in Fig. 48. The top of this stretcher should

be low enough to clear the largest pulley, and high enough to clear the hat of your tallest man. You would perhaps find it convenient to go between the spokes of a large pulley.

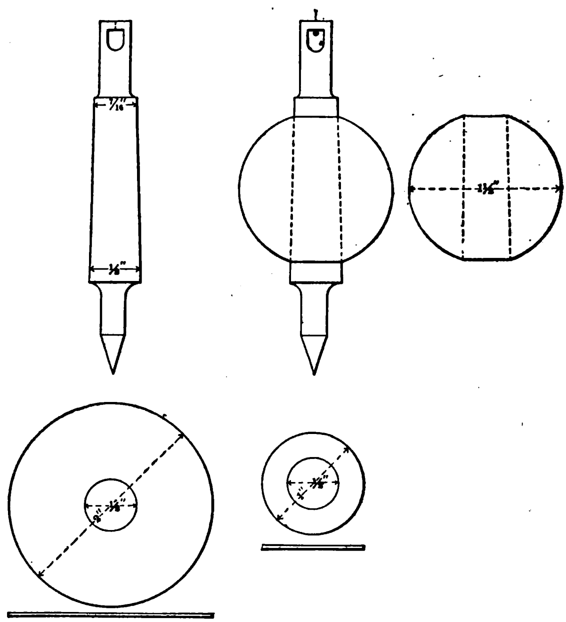


Fig. 47.

Now having located your stretcher, find approximately the position of your line, and drive a nail a foot or more below it in a vertical line, and another nail anywhere for convenient winding. The advantage of this plan is that the line can be easily adjusted as it

merely passes over the stretcher, and is free to respond to movement either way; then when the final adjustment is made, and is ready for its final stretch, it is only necessary to pinch the line to the nail with one hand, while the other is at liberty to unwind, stretch and rewind the line without fear of its shifting.

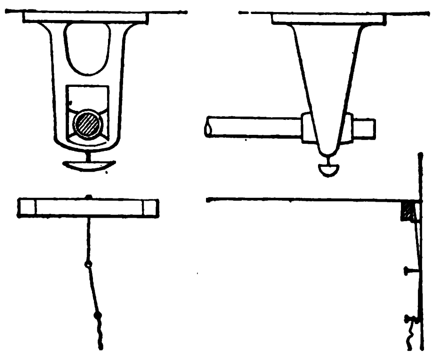


FIG. 48.

The line being adjusted over the stretchers, we will now proceed to set it. Begin at the $2\frac{7}{16}$-inch end, by throwing your plumb line over the shaft and setting your line at that end, right with the *center point* of your bob. Having done so, go to the other or $3\frac{15}{16}$ end of your line, and set the line so that the edge of the *ball* of your bob just touches it. Now go back to the $2\frac{7}{16}$ end and see that the necessary adjustment did not alter it. Having proved this, give your line the final stretch and try if it is right at both ends. You now have a center line (though the edge instead of the center of the

shaft is used) that may remain up for days if necessary without fear of disturbance.

It is best to go over the whole line first, before disturbing anything; so starting at the first hanger at the $2\frac{7}{16}$-inch end, throw your plumb line over the shaft, and record on the floor in chalk beneath it whether it is O. K. or wants to go either way, and how much; then go to the next hanger, and so on to the end. A short study of the conditions enables one to correct the faults, with a knowledge of the requirements, and consequently in the least time and with the least trouble.

Now suppose we start at the $2\frac{7}{16}$-inch end to inspect the line, we use the center point of the bob on the line so long as we are testing $2\frac{7}{16}$ inches.

When we get to the $2\frac{15}{16}$-inch part, which is $\frac{1}{2}$ inch larger, we use the half diameter of the stem, the edge of which should just touch the line.

When we come to the $3\frac{7}{16}$-inch part, 1 inch larger than $2\frac{7}{16}$, we use the 1-inch disk, slip it on to the stem, and when it just touches the line with its edge it is O. K.

The $3\frac{15}{16}$-inch, being $1\frac{1}{2}$ inches larger than the $2\frac{7}{16}$-inch, will be right when the ball of the bob is in light contact with the line.

The 2-inch disk would be suitable for the next size, and other disks or modifications of the bob proper might be made to suit circumstances.

Now having straightened the line, the next process is to level it. As in some cases your pulleys will be too close to place your level where you want, make a

light iron frame as per Fig. 49, making the suspending members of sufficient length to admit of your reading the level conveniently when standing on the floor. Hang your frame on the shaft, and put your level on the straight-edge below; in this way travel along the shaft, placing your frame where convenient. Be sure that one end of your frame does not rest on a shaft of different diameter, a key, keyseat, or anything to distort the reading.

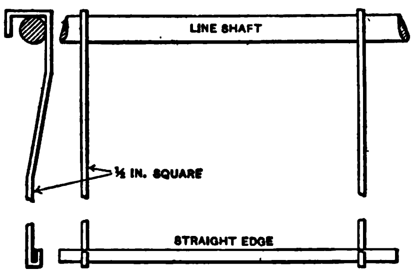


FIG. 49.

Never be content with trying your level, especially an adjusting level, one way; always reverse it and try again; for if it is out of truth at the start, you might want to go through the roof or down cellar at the finish. Get into a habit of reversing your level, and so prove your work as you proceed.

V

APPARATUS FOR LEVELING AND LINING SHAFTING

THE first apparatus explained in this chapter was designed by the late Chas. A. Bauer, and is a highly perfected instrument.

For those who have lined and leveled shafting with an engineer's transit and level it is unnecessary to say anything of the advantages of that method over the cruder methods usually employed. It is not only done much more rapidly and economically, but the greater accuracy with which the work is done goes on paying dividends in decreased friction and loss of power and in lessening of wear.

The apparatus we now illustrate (Fig. 50) has at the top a hook, which is passed over the shaft, as indicated; on the straight portion of this hook are two sliding jaws which are so set that the shaft will just pass between them. Set into the face of this hook is a commercial 6-inch steel rule which facilitates the setting of the jaws, and they are of course so set that the tubular portion of the hook or leveling rod is centered vertically under the shaft. Within the outer tube, which is about 1 inch outside diameter and nicely japanned, is another tube, and inside this a third tube,

FIG. 50.

these being arranged *à la* telescope slide, and clamps being provided so that the length or distance from the shafting to the target may be anything desired from 4 to about 10 feet. At the lower end of the third or inner tube is a swiveling head to which the target is attached, and nurled nuts at this point give means of adjusting the sighting point of the target to the exact hight of the transit or level sighting line.

The target is a brass plate 5½ inches diameter, on the face of which is a recess milled for the reception of a second commercial steel rule, which in this case is vertical and can be moved vertically and clamped in any desired position with reference to a line drawn upon the target. At the center of this scale is a very small hole through which the light of a hand flash lamp may shine to form the sighting point. The slot through the target at the right of the scale is provided with a single thickness of white cloth, which permits enough light to pass through it to help in finding the target in the field of the telescope.

The object of providing a vertical adjustment for the rule on the target is so that when passing from one diameter of shafting to another in the same line, as sometimes happens, the scale can be moved up or down just half the difference of diameter and the sighting point thus be kept at a constant hight.

The target is readily detached from the rod, and may then be placed upon the small standard (Fig. 51) which has at its base a V adapted to go over the shaft. The standard is tubular and the wire (about ⅛ inch diameter) may be adjusted and clamped at the desired hight. The

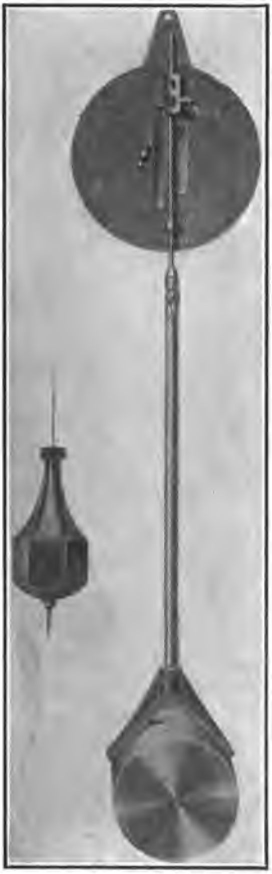


FIG. 51.

target fits over the wire as shown (rear view of target) for leveling lines of shafting that may be near the floor, or, with the target removed, the V and wire form a sort of length gage or caliper with which the shaft may be made parallel to a line or wire stretched at the side of it. Two different lengths of wire are provided for this purpose.

The plumb-bob shown is part of the equipment and is a very superior article. A new feature it possesses is in having its larger portion hexagonal instead of round, so when laid down upon a plank or scaffolding it will lie there instead of promptly rolling off and falling to the floor. The entire apparatus is, we think, very well designed for its purpose.

Tool for Leveling Shafting

The instrument shown in Fig. 52 is a good one for use in leveling up shafting. It can be made to fit several sizes of shaft, or all the sizes ordinarily found in a factory.

When the instrument is placed on any piece of shaft and leveled up with the attached level, the plumb line will hang exactly the same distance from the shaft center every time. In this case the distance of line from center is 6 inches.

A handy apparatus for use in leveling up long lines of shaft can be made as follows.

Take two pieces of finished material, fasten together as in Fig. 53 and cut out as shown at *A* and *B* in Fig. 54. The opening *A* is made so that the piece can be hung over the shaft, and the opening *B* is made for the reception of a wooden straight-edge.

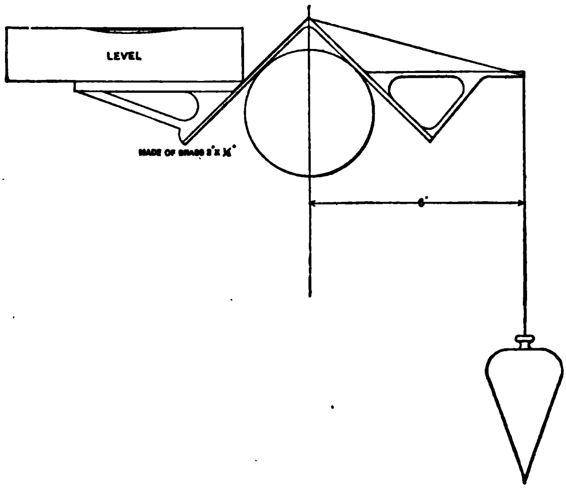


FIG. 52.

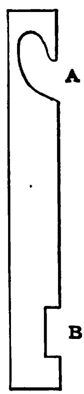


FIG. 53. FIG. 54.

Make the straight-edge out of $1\frac{1}{4}$-inch stuff. Be sure that the edges are parallel, the width just enough less than the width of opening *B*, Fig. 55, to enter it,

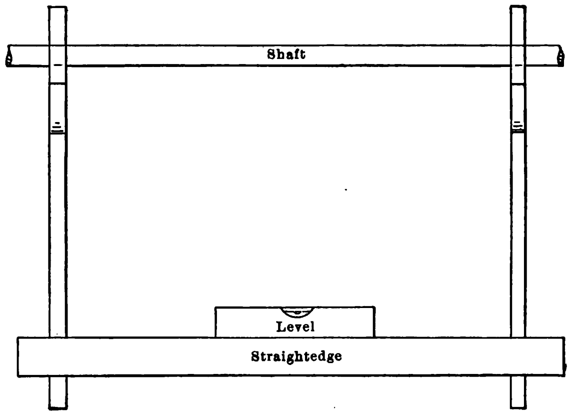


FIG. 55.

and the length 6 or 8 feet, to suit convenience. Use the apparatus with a level, as in Fig. 55, taking care that the suspension pieces are always on the same size shaft.

VI

SOME PRACTICAL KINKS[1]

A PULLEY on one of the motors at a certain plant had been giving some trouble by becoming loose and working its way along the shaft toward the motor bearing. Each time the pulley became loose, the set-screw was loosened, the pulley put back in position, the set-screw made tight and the motor started. After a few trials it was found that this would not prevent the pulley from working its way along the shaft. In order to overcome this difficulty the pulley was placed

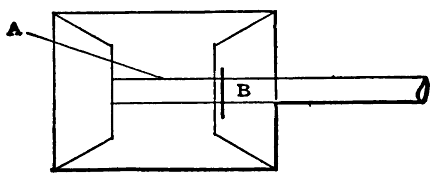


FIG. 56.

in its proper position, a line was drawn around the shaft close to the hub and, after the line was scribed, the pulley was removed and the shaft was burred upon the line as shown at *B*, Fig. 56. The pulley was then put back and driven close up to the burred line, the

[1] Contributed to Power by Wm. Kavanagh.

set-screw made tight and the pulley is now running without any apparent tendency to travel from its proper position. It will be seen that the position of the set-screw as indicated by the line at *A* is a poor one and calculated to give plenty of trouble at the most inopportune time.

Not long ago a cast-iron pulley had to move along a countershaft in order to make room for a pulley of another diameter. The pulley had not been on the shaft long, so it was thought that little work would be required to move it. A heavy bar was placed against the hub and a sledge hammer was used to strike the bar. After an hour and a half of heavy work the pulley was not moved over 1 inch (it had to be moved 16 inches), so it was suggested that a Bunsen burner be attached to a gas pipe by means of a hose and placed beneath the hub. The plan was immediately adopted. The burner was placed beneath the hub, the gas lit and allowed to heat the hub. After about twenty-five minutes it was found that a blow from the bar was sufficient to move the pulley. The pulley was moved the 16 inches inside of twenty minutes.

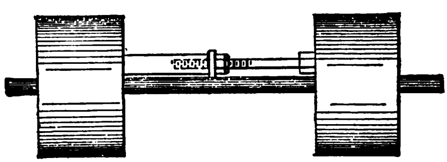


FIG. 57.

A very handy arrangement for moving pulleys is a bolt and nut. Fig. 57 shows the bolt and nut with

a piece of pipe attached. A piece of pipe can be cut to suit the distance between the nut and hub of one pulley while the bolt head is against the other hub. The nut is screwed back upon the bolt as far as possible. A washer is then placed against the nut, and a piece of pipe cut to suit. Of course, the pipe must be large enough in diameter to fit over the bolt. If we screw back upon the nut, a powerful strain can be brought to bear between the hubs and in all probability the pulley will move.

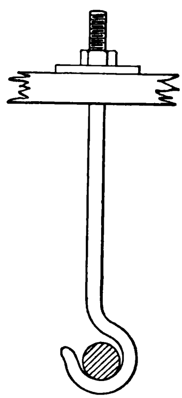


FIG. 58.

In taking down solid pulleys from main or counter shafting it sometimes happens that a hanger must be removed to permit the pulley to be taken off. A first-rate plan is to make a couple of long bolts hooked at the end as shown in Fig 58; pass the hook around the shaft and the threaded end through a hole in the stringer. By screwing up the nut as shown, the shaft

and remaining pulleys can be kept in position, obviating the use of tackle, not to mention the labor required to hoist back the shaft into position. The application of this contrivance is especially valuable where heavy cone pulleys are required to be lowered or changed. It will be seen that if we employ a pipe thread we will be enabled to suit almost any condition of length that may arise between the shaft and stringer.

VII

PRACTICAL METHODS OF LOOSENING PULLEYS

WHEN a solid pulley is to be removed from a piece of shaft for any reason, it is not good policy to use sledge hammers on the spokes or hub to do it. Cast iron in pulleys is too liable to break or crack under repeated blows.

In Fig. 59 one ready method is illustrated by which the pulley may be removed. When a place between two walls can be found that will admit of this arrangement, proceed as shown to force the shaft through the pulley, substituting longer pieces of pipe as the shaft is forced through farther.

In one case where a large pulley was stuck on a 7-inch shaft and its removal was imperative, the shaft was sawed off (with large hack-saws) close up to the pulley hub and two $\frac{5}{8}$-inch holes were drilled into the shaft parallel to its axis, as shown in Fig. 60. These holes were drilled so that they were 90 degrees apart and came within $\frac{1}{16}$ inch of the hub of the pulley. The hub was 14 inches through and these holes were 8 inches deep; but that was enough to loosen up the shaft so that when the pulley was laid over on beams with

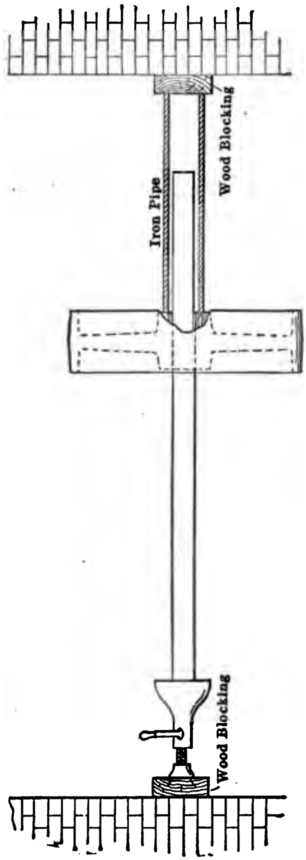


FIG. 59.

the shaft hanging through, a sledge hammer applied on the shaft end soon drove it out.

Another way to remove a pulley is shown in Fig. 61, where a ram is used. The ram is another piece of old shaft. To prevent its damaging the pulley hub and also to have its force applied most advantageously, it should be used in a direct line with the direction of removal. To do this, the method shown in Fig. 61 is self-explanatory.

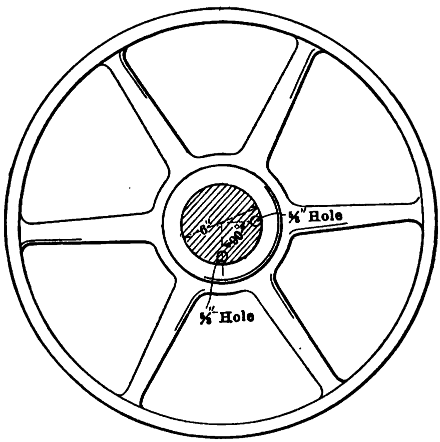


Fig. 60.

Another good method of removing an obdurate pulley is illustrated in Fig. 62, where the bolts *W*, *W* must have long threads and the work is done by pulling up on the nuts *A*, *A*. This method can be used only when the end of the shaft can be reached and used as shown. In using this method, care must be exercised

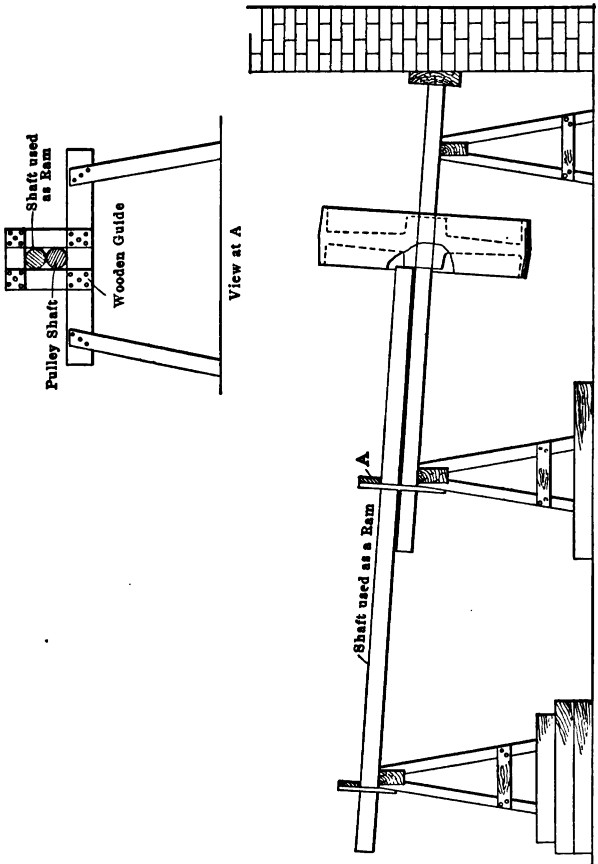

Shaft used as Ram
Pulley Shaft
Wooden Guide
View at A
A
Shaft used as a Ram

FIG. 61.

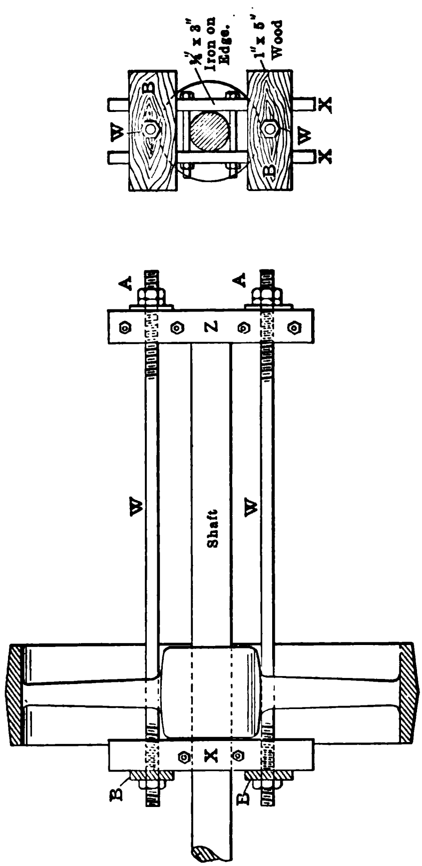


FIG. 62.

in the pulling up on the bolts *W, W*, keeping the strain equally divided between the two by pulling a little at a time on each.

If the pulley comes extra hard, it can be assisted when the strain is on the bolts by striking at *X* with a sledge.

A good device for removing motor and generator pulleys that are near the shaft end is shown in Fig. 63. The arms *Z, Z* are adjustable to take hold of hub or arms, and the screw applied to the shaft center will do the rest.

To run a pulley off a shaft without injury to the hands, use a monkey wrench on the rim of each pulley, as shown in Fig. 64. One pulley on the shaft can be selected for a hold-back; one monkey wrench there will hold the shaft from turning, while the other will turn around the shaft the pulley which it is intended to remove.

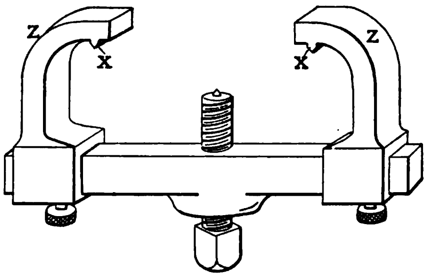


FIG. 63.

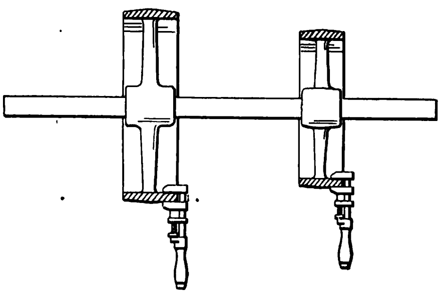


FIG. 64.

VIII

SPLICING LEATHER BELTS[1]

THE first thing is the tools for the different kinds of work. These may be usually changed somewhat to suit the taste of the user, but in the main the style and kind herein shown in attached drawings cannot be very much improved upon.

Figs. 65 and 66 show a splice opener for heavy belts. It is made of ½-inch tool steel with the point spread out about 2 inches wide and well tempered, after which it is ground to a good sharp edge, and then an oil stone run over the edge until it has been dulled so that it will not cut. The right kind of an edge can only be secured by trying; it is one of the tools that is very hard to get just right. You will notice that the manner in which this splitter is built may seem to be rather too much work to bestow on such a simple tool, but the reasons for so doing are as follows: in opening a 36-inch belt an old splice opener that was driven into the handle like an ordinary file was used and the handle split; that sharp point came back through the handle, and when it finally stopped it had gone about 2 inches into the palm of the operator's hand. Some ½-inch hexagon steel was turned down 6 inches, just

[1] Contributed to Power by Walter E. Dixon, M. E.

FIG. 66.

FIG. 65.

enough to round it up; then a solid brass washer was turned out 1¼ inches in diameter and 1 inch thick, a hole bored through it that was a driving fit on the piece of steel and was driven down to the shoulder. Washers were cut out of old pieces of belt and put on with a liberal coat of glue on both sides; when the handle was filled, a steel washer which was ½ inch thick was screwed down hard on the leather washers, and when it had dried well the whole was turned down to size shown in the sketch. Two of these tools were made, one for belts up to 18 inches, and another that will reach through a 40-inch belt. The tool shown in

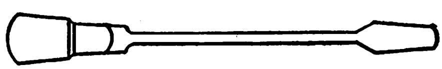


FIG. 67.

Fig. 67 is an ordinary heavy screwdriver with the point rounded nicely, and it is used to raise the thin points that the larger tool will sometimes tear.

Fig. 68 shows a handle made almost like the one in

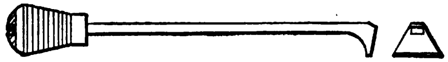


FIG. 68.

Fig. 65, with the exception that the brass washer referred to in Fig. 65 is here turned down to ¾ inch, commencing ½ inch from the large end, which is 1 inch in diameter. The leather washers are slipped on over the small part until it is filled, and then a washer is screwed on the small end and the whole turned as shown

in the sketch. A hole that will tap out $\frac{3}{8}$ inch is bored in the large end of the brass center, and then tools made with threaded ends on them that will fit into it. These tools are made of $\frac{3}{8}$-inch tool steel with scraping ends, as shown. These scrapers are used only for removing glue that is too hard and too thick to be removed by the scraper shown in Fig. 69.

Figs. 69, 69a and 69b show views of the only tool

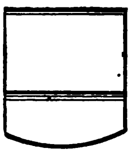


FIG. 69.

FIG. 69a.

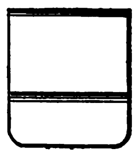


FIG. 69b.

that is hardly worth being referred to as a leather-cutting tool. It is made of a thin piece of steel, about 18 gage, or any old hand-saw will make the very best scrapers that can be secured. They should be about 4 inches square, perhaps a little smaller, and fixed in a hardwood handle (usually of hard maple), simply by sawing about $2\frac{1}{2}$ inches into the handle and then driving the blade in. The saw cut should be just a trifle thinner than the piece of steel. Should they get loose from use, a piece of paper folded over the back of the blade and forced back into the handle with the blade will usually tighten it all right.

This is the tool that will ordinarily worry the novice more than all the rest to keep in proper condition. Fig. 70 shows an exaggerated view of how the blade

should look when properly finished. It should be hooked considerably.

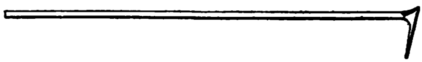


FIG. 70.

Fig. 71 shows a small steel for sharpening the scraper after it is turned, and it should be absolutely smooth.

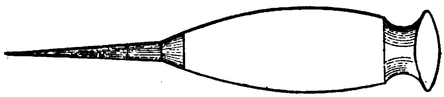


FIG. 71.

Fig. 72 shows the equipment for turning the edge of the scrapers. A large three-cornered file, about 12

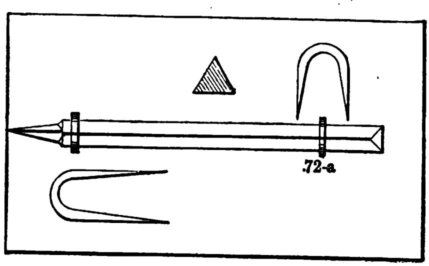


FIG. 72.

inches long, which has all the teeth ground carefully off of it and then nicely polished, is fastened to a piece of good clean belt leather by means of the staples shown.

Fig. 73 shows the method employed in turning the edge of the scraper, which is as follows: After the blade has been set firmly in the handle, grind the edge rounding, as is shown in Fig. 69; then grind sharp with a good long taper of about $\frac{3}{8}$, and grind from both sides just as you would an ordinary axe. After you have a good smooth edge on it, put it on an oil or water stone and put as fine an edge as possible on it, then put on a smooth piece of leather and hone it down until it would shave you. You will then have a tool that will do a world of work for you, "if you will turn it right."

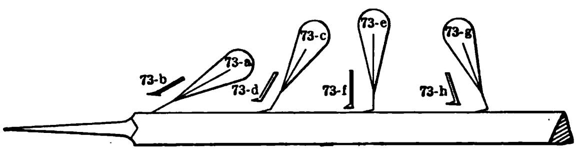


FIG. 73.

The method shown in Fig. 73, if properly carried out, will do the trick for you; the thing to be remembered is that at no time in the turning of the scraper must the cutting edge bear on the smooth file. The first position is not shown right; the handle should be allowed to touch the file the first few times it is passed over, and then gradually raise the handle and keep on passing the blade from side to side, as is shown in Fig. 74, allowing it to slip off on the leather every time you cross the file; this is to keep the corners in proper shape. Another thing to remember is to bear down on the blade as it is passed over the file; you can't bear too hard; the only thing to look out for is not to raise

the handle too fast. An ordinary blade can be turned in about fifty strokes across the file. The edge turned over should be at least $\frac{1}{16}$ inch long and should be well hooked, as is shown in Fig. 73.

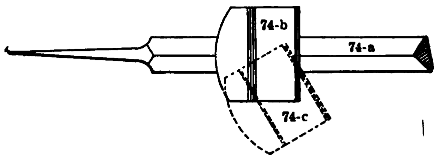


FIG. 74.

It is well to keep on hand about six of these scrapers, and as they get too dull to cut leather use them on glue. With one good scraper that is not too sharp all the glue can be cleaned off of both points of a 36-inch belt in from five to ten minutes. When the edge gets a trifle dull, use the small steel on both sides of the edge; first wet the steel with the lips, it makes a much better edge. For the benefit of beginners who may attempt to splice a belt for the first time, do not use a glue that will not allow you to remove the clamps and put on the full load in forty-five minutes after the glue has been applied and well rubbed down. The time given here applies only to clean belts that are absolutely free from all oils, and does not include old oil-soaked leather that no glue will ever dry on.

Fig. 75 shows the equipment necessary to do a good, quick job on a belt, and most of them are required to be done quickly and well. With such an outfit and half-dozen sharp scrapers a joint in a 36-inch belt can

be made and run again in four hours after the engine is stopped. This includes all the time consumed in putting on and taking off the clamps, etc.

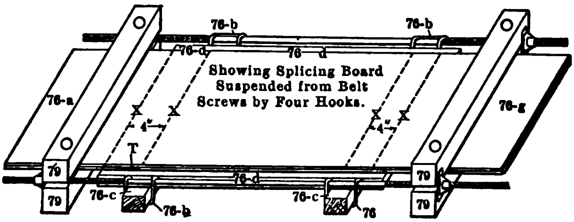


FIG. 75.

The top of the platform, 76d, is level with the bottom of the belt and is held in position by the hooks, 76b, which are shown in Figs. 75 and 76. These hooks slip over the 2x4-inch pieces that project outside the platform to which they are attached, and should be made of three-quarter iron and not too long, or some difficulty may be experienced in getting them on the two-by-fours.

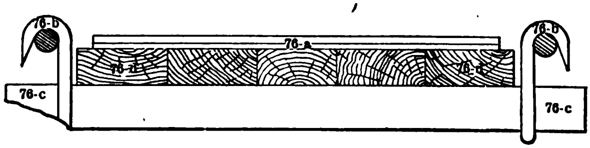


FIG. 76.

The rods should be long enough to take care of the longest possible splice and still give plenty of room to work. There should be about 2½ feet between the

inside ends of the threads and the threaded end should be 3 feet long. This will make the rod 8 feet 6 inches long, and it will be none too long at that. For instance, in removing the glue from the splice, if the last end point is very close to the clamp, there will be great difficulty in cleaning it and also in fitting the leather after the belt has been shortened. What is meant by

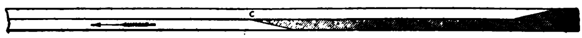


FIG. 77.

the head end splice is the one that is on the pulley first — the arrows in Figs. 77 and 78 will make this clear: they indicate the direction in which the belt should run; therefore that end of the piece of leather

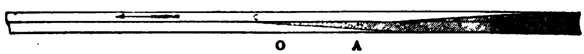


FIG. 78.

that is on the pulley first is the head end (or first end) and the end that leaves the pulley last is the last end. If the two belts shown in the sketch were reversed, the points would be turned up by everything that touched them; whereas, running in the direction that they do, everything that touches them has a tendency to rub them down.

We will suppose that the belt shown in Fig. 75 had a "first end" point that opened on the top of the belt instead of the bottom as this one does (see left-hand end of belt between the clamps, on the lower side); one can easily see how hard it would be to work if the clamp were near the point. There should always be enough

room between the clamps to allow the splicer to take the last end (which is always the forked end), carry it entirely over the clamp toward the left in Fig. 75, lay it down on that part of the belt that is outside the clamp and slip an extra splicing board under it. Fasten the two belts and splicing board all together by means of a couple of 8-inch hand-screws (of which every belt splicer should have at least six or eight); then clean and shape it to suit the other end. It can be passed back over the clamp from time to time and tried for a fit.

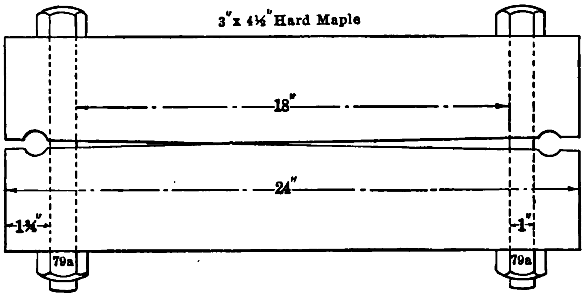


FIG. 79.

The proper mode of procedure in splicing a belt on the pulleys is as follows: Decide on where the belt is to be opened, and always open it in the worst place in the belt for that is the place you certainly want to fix. Pay no attention whatever to any former splicing place that may be in the belt, but take it apart at any place where you are sure repairs are actually necessary. First put in the most convenient place possible the point that you have decided to open and then put the

clamps in position. If you are sure that it is going to require very hard pulling to get it as tight as you wish, take a damp cloth, moisten the inside of the clamps and then sprinkle powdered resin on both upper and lower clamp. Put the "first end" clamp on first, as this is always the easiest point to clean and fit; decide how much you will have to take out, or as near as possible, measure off this amount on the belt and place the clamp this distance plus about 10 inches from the "first end" point. This extra 10 inches will give you plenty of room to clean the glue off and also to shorten up the belt the right amount, for all the shortening must be done on the "first end" point on account of the ease with which the new scarf can be made.

Should you try to shorten up from the "last end" point, by referring to Fig. 78, you can easily see the amount of work you would be in for. There would be two thin ends to scarf, and outside ends at that; whereas if you shorten up from the "first end" you make only one thin end and that one in the inside of the belt.

The first clamp, with the center mark of the clamp coinciding with the center of the belt, should be very tight; for should it slip when the load is put on, it will very probably slip in the middle of the belt and may not slip on the edges at all. Should you glue it in this condition, the chances are very much in favor of the outside edges giving away on a heavy load, due to the middle being too long. After the first clamp is in position and tightened, put on the second one and leave the bolts loose, so that it can be slipped easily. Then put the belt rods in position with just a "full

nut" on each end and tighten the clamp. Tighten the rods enough to take most of the load, then get the large splitter shown in Figs. 65 and 66 and open the joint. The place to commence is between *X X* in Fig. 75; this inclined point is about 4 inches long and must be opened at both ends of the splice before the middle is touched.

The tool should be entered at *O*, in Fig. 78, and worked gradually toward *A*; when the point is raised to *A* clear across the belt, open on down to *C*. After both ends of the splice have been opened up in this way, proceed to open the middle, which is now an easy task, there being no thin stock that a separating tool will pass through easily. After the belt is entirely apart tighten up on the rods until the belt is the proper tension and hang the hooks (76b, Fig. 75) on the belt rods. Throw the two ends of the belt back over the clamp and put the splicing board in position. After this is in place, throw the two ends of the belt back on the board and proceed to lay off the scarfs. To do this, first take a square and get the two thin points perfectly square, then put the "first end" point in between them. This is shown very clearly in Fig. 77, the shaded end being the last end. Of course the "first end" point at *C*, Fig. 77, will have to be cut off before the belt will lie down properly; the amount to cut off of this end will be just as much as you have shortened the distance between the clamps. After the point has been cut to the right length, take the square and make a mark across the belt, using the end of the thin point as your measure for length; then without moving the

belt make a mark on the edge of the belt, showing just where the lower thin point came on the bottom. Throw the "last end" over the left-hand clamp out of the way and scarf down the top of the "first end" point, letting the scarf be about 4 inches long. Be careful not to gouge a hole in the belt where the scarf is started, but try to make the inclined plane from *X* to *X* perfect; try to keep the whole surface of this incline true and straight. After the short 4-inch scarf is finished, clean the glue off of the inside of the "first end"; lap up to where it enters the "last end"; then turn it over by bringing it over the right-hand clamp, place a scarfing board under it and make the scarf shown at *T*, Fig. 75. Now clean all glue off the "last end" lap and take a sharp scraper like the one shown in Fig. 69 or 69b, place a piece of glass under the points that have been previously squared up, and scarf them down to a knife-edge.

After the thin points are properly scarfed, lay the whole splice back on the splicing board just as it will be when it is glued, and do any fitting that may be necessary. Be very careful to get it thin enough, or it will make a hammering noise when going over the pulleys. When scarfing down the thin points with the scrapers, be sure that they are very sharp; if not, they will tear the point off when it gets down to an edge; also give the blade a drawing motion in order to facilitate cutting. It may seem to the novice that to use a piece of glass to scarf on, when one is using a tool with a razor edge, is a trifle inconsistent, but it is not so in the least; if the blade is held well back at the

top and a considerable pressure applied to it, there will be no danger in the edge actually touching the glass; the edge is turned past a right-angular position, or hooked, and the heel is all that touches the glass. A good piece of plate glass about 12 × 18 inches is large enough for any width of belt, although a piece much smaller will do all right. Do not attempt to do any scarfing on the board 76d, for if you do it will be so full of holes that have been gouged by the scraper that it will be ruined for any purpose.

This board must be kept smooth in order to be able to do a good job of rubbing down when gluing. Never hammer a glue joint in order to set it; it is just that much unnecessary work and does absolutely no good; simply get a smooth block of wood 2 × 6 × 8 inches and rub hard and fast as soon as the glue is applied. Do not try to glue more than 6 inches in length at one time. Use a heavy brush — a high-priced paint brush is the best; the regular glue brush is about the only thing in existence that will not put on any glue at all — about a 3-inch brush is the thing; have the glue just as hot as it is possible to get it. Keep the brush in the pot all the time the glue is heating; also have a strong stick made somewhat like a three-cornered file, only larger, in the glue — this last is used to scrape off the brush all the glue that it is possible to get off without allowing the glue to get too cold. When you take the brush out of the pot, work fast; get all the glue possible off the brush and get the rest on the belt at once. Make two or three fast strokes across the belt and close down the splice and rub for dear life. After the first brushful

has been applied (and rubbed for about two minutes), have an assistant raise the point up until you can see the glue breaking all across the whole width of the belt. Then have a second brush ready and repeat the former process, with the exception that you need not apply the glue to both sides of the leather as in the first case; for if you will keep the brush down in the fork between the two laps you will give both sides a coat, and in addition to the time saved by using this method you will get the joint closed while the glue is hot. As fast as you go across the belt with the brush, have the assistant roll the belt together after you; when you have used all the glue out of the brush, the joint is closed and ready to rub. You will keep the glue much hotter by immediately closing the splice after the brush, and there is nothing else so important as using hot glue; as soon as it commences to get shiny on the surface the thing is all off and it will not hold anything.

You cannot do any quick work with water in your glue — that is, unless it is old and has been heated up several times. If this is the case, it will have to be thinned with water. The proper consistency is about that of a very heavy grade of cylinder oil; if it is too thin, it will not dry in any reasonable time and it will also cause pockets in the splice by opening up after the joint has been rubbed, and the air in the pockets will open the whole splice. In important work never use a glue that will not stick so tightly between every application belt that after rubbing down you can give it a good, hard pull without its opening up. In all statements regarding the time necessary for the joint to dry,

the belts are considered absolutely clean, dry and free from all oils.

The most disagreeable portion of the belt repairer's work is the splicing and repairing of oil-soaked belts. It is a well-known fact that the action of oil and that of glue are in direct opposition to each other: the oil prevents sticking and the glue sticks, if it has a chance. Such being the case, the first thing to do is to eliminate the oil completely, and the efficiency of your joint will be in direct proportion to your success in getting rid of the oil. To this end secure a large gasoline blow torch, such as painters use to burn off old paint. If you are not used to it, be very careful; at all events, have a bucket of dry sand to use in case of trouble. Just throw the sand on the fire and the fire will go out — that is, if you can get the sand in the right place.

The torch is to be used after the splice has been all completed except the thin points. The flame will burn them if finished, so leave them tolerably thick until after the oil has been removed; then finish them as directed before. When the scarfs have been made and the old glue has been removed, turn the flame (which should be an almost invisible blue if the torch is working properly) directly on the leather and move it over all the surface of the splice until the leather has become thoroughly heated; never allow the flame to remain directed at any point long enough to make the oil in the leather boil. If you do, the belt is burned. Continue to move the flame over the surface of the belt until the leather is so hot that the hand can scarcely be held on it. With one of the scrapers shown in Figs.

69 and 69b (69b preferred) scrape the oil off as the heat raises it up. Turn the cutting edge of the scraper up and wipe the oil off after every stroke; keep the scraping process going right on after the torch; never allow the leather to cool off until you can get practically no oil and the leather begins to turn brown. By heating the leather and bringing the oil to the surface you do just what the glue does when you put it on an oil-soaked belt without removing the oil. By means of the heat contained in it, it brings up all the oil near the surface to which it is applied and in consequence does not take any hold on the leather.

It will take two men with all the necessary tools and appliances at least six hours of good hard work to remove the oil from a well-soaked 36-inch belt — that is, to remove it to an extent sufficient to warrant the gluing of it.

In case of overflows in which the wheel pits are liable to be filled with water, pour cylinder oil on all belts that are liable to get wet and then remove them from pulleys if they will be covered for more than twenty-four hours, clean them with gasoline and they will be found to be all right and dry.

Hold a clean piece of waste against all belts at least twice every twenty hours, and wipe them clean.

IX

THE CARE AND MANAGEMENT OF LEATHER BELTS[1]

OUTSIDE of the direct care and management of high-pressure boilers and the steam lines pertaining thereto, there is no other part of a power or lighting plant, mill or factory in which a large number of indirect connected machines are used that is of such vital importance as leather belting and rope drives. The subject under discussion in this chapter will be the former, and the selection, care and management thereof.

The first thing in order will be the selection of a leather belt, and when we consider that all makers make good belts, that there are no particular secrets in the belt-making business, and that in order to get the very best we must take every advantage of all small details in construction, it stands every engineer and belt user in hand to get all the information available; for we must remember that the percentage of good hides does not run very high, that all that are bought go into belt stock of some kind or other, and that some one must buy the goods that are not quite up to the standard of belt excellence. It is very evident that no man wants anything but the best when he is paying

[1] Contributed to Power by Walter E. Dixon, M. E.

for the best, and it is also evident that no maker is going to say that he makes inferior goods; so therefore we must read the quality by what is in sight, and in the judging of leather that is already made up, the proposition resolves itself into a very hard one.

The two principal things left for an opinion to be based upon as to quality are the relation the pieces that constitute the laps bear to the hide from which they were cut. They should, in belts running from 18 to 36 inches, be cut from the center of the hides, or should be what is known as "center stock." Of course all belts should be "center stock," but where they are very narrow or so wide that one hide will not be wide enough to make a lap, then there is always a lot of narrow stock worked in that cannot always be strictly center.. The next thing to look out for is brands that are so deep that they destroy the life of the leather and will cause it to break after being used. Then look out for the length of lap. If this is too long, you will know that it runs into the neck, for about all that it is possible to get out of average hides and still leave nothing in that is not first class is 54 or 56 inches. Ordinarily, you can tell if a lap is "center stock" by the marks that run down either side of the back bone; they will be usually a little darker than the rest of the belt. These marks or streaks should be in the center of the belt. The principal objection to neck leather is that it is liable to stretch excessively, and on this account it will put too much load on the piece immediately opposite it in a double-ply belt; for the point of one side is in the middle of the lap on the other side. Next look out for

holes, which will usually be found so nicely plugged as to escape detection unless subjected to the most careful examination.

Next in importance is to buy a belt that has already been filled with some good waterproof dressing. It is quite likely that to buy a belt that has been filled means to buy one that perhaps has some bad leather in it that would be seen in a dry oak tan belt, and also that the adhesive power of the filled belt is not quite equal to the dry one; but the points that the filled one possesses over the one not filled are, first and mainly, "it is filled when you buy it with a preparation that does not injure the leather in the least," and the preparation you will fill it with, for it will be filled, will be engine oil and water, a combination that will ruin any belt made and also get it in six months into a condition that will make a permanent repair with glue impossible, for machine oil and moisture are strangers to glue and will ever be. More good belts are ruined by being soaked with engine oil until the points come loose and then pulled out of shape than from any other cause. Of course *you* may be able to keep a main engine belt that runs through a damp wheel pit and basement, and through a long damp tunnel to a main driven pulley that has two big boxes that are just as close to the pulley as a first-class machine designer could put them, and never get a drop of oil or water on it. But this is not likely.

One very common cause of trouble with engine belts is the fact that such belts usually run under the floor, where there is considerable moisture. Then the oil

table under the average large Corliss engine will leak around dash-pots and rocker-arm shafts, and some oil will fly from the eccentric oil cups, get into the wheel, run around the rim and get to the belt; if the belt is not filled a very few drops of oil will make a large spot on it. Then, if an engine does not run the whole twenty-four hours, while it is off, watch. A few drops of water from a leaky valve stem whose bonnet drain is stopped up, as it will sometimes be, has a way of getting through the floor and falling on to the belt and running down the inclined inside of it until it finally comes to the flywheel, which, with the assistance of its crowning face, very kindly makes a nice pocket for said water and proceeds to drink it up. Result: the glue is loosened and the belt may come apart in consequence. Should there chance to be a point just at the bottom of this pocket, it will get the glue soft enough to slip but may not open up, which is much worse than if it did open up; for it may slip away from the shoulder of the splice for half an inch, and when the engine is put to work it may close down by running under the wheel and stick. If it does, the result is that at no very distant day you will find a break at that particular place, right across the face of the belt. The reason is that the load was all taken off the inside half of the belt by point slipping, thereby making the inside of the belt too long and putting all the load on the outside. The outside will continue to do all the work until it stretches enough to bring the inside back into service again. During this week or month you have been pulling your load with a single belt, not a double one,

and after a short time you will find the break referred to above in the shape of a clean, well-defined crack extending across the belt parallel with the points of the laps. Now of course you are going to send for the man who sold you the belt and ask him to fix it. If he is a wise man and understands his business, he won't do a thing but show you right under that crack a point that does not come up to where it should come. Then the thing for you to do is to say to him that the belt is examined every time it is put into service and that you have noticed that the points he refers to all come loose during a "run," that any one knows that a few drops of water would not take any belt to pieces while it was running, and if it was water, why did it not take it apart everywhere, etc? And finally crush him completely by telling him that your men have no time to put a pair of clamps on a belt in order to pull back into its proper position every point that comes loose; that if they did do it they would have no time for anything else, especially in the present case, and that if his people had made the belt right the glue would have held, anyway.

After he has given you a new belt or repaired your old one, just take my advice and box that flywheel up above the top of the eccentric oil cup, at least 12 inches, and get some good, heavy tin or zinc and put a tight roof over the belt, under the floor.

First put in a ridge pole out of 1½-inch pipe, starting at the face of the wheel and running in the direction of the main driven pulley, holding it firmly in place at each end with a strong iron clamp. Then solder into

each edge of the strip of tin, which should be long enough to reach beyond any possible leak through the floor or oil table, a piece of ½-inch pipe, and put the tin over the ridge pole with a piece of small pipe on either side. Ordinarily the belt goes out past the cylinder; if it runs through a bricked-up runway on its route to the main driven pulley, just fasten the two pieces of ½-inch pipe to either wall and have the ridge about 6 inches higher than the outside ones. Then every drop of oil or water that comes through the floor will fall on to the roof and run down to the walls and be carried down to the floor of the pit and have no chance to touch the belt.

One of the most difficult things in the operation of large stations where a large number of belts are used is to keep them thoroughly clean and free from moisture and machine oil, the latter especially. One very hard problem that confronts all designers of machinery is the prevention of oil leakage from boxes. In several plants with as many as six dynamos of the same kind and the same design, at least four of the six have leaked oil every time they were run. The others did not leak as a usual thing, and all were equipped with the most modern methods of holding oil.

Now we come to the building of the belt, and we will notice only such points as interest the engineer or buyer. The first thing is to see that the laps are of uniform thickness, so that the belt will run quietly; and it should be absolutely straight when unrolled on the floor. If it has a long, graceful curve in it, look out; for it will not run straight on the pulleys until it has stretched

straight, and by that time one of its edges may be ruined by coming in contact with the floor or some other obstacle. Next notice how long the leather is from which it is made. It should not show more than 52 inches, and then there will be 4 inches hidden by the point that is out of sight. Then see that the joints are broken properly. For instance, find the center of any piece of leather on one side of the belt, and then look on the opposite side and see if the joint is right under your center mark. It should be by all means, for right here lies the most important thing about the construction of leather belts. A belt whose laps are all the same length, and which has all its joints broken correctly, will put the same load on the glue throughout, and that is what must be done in order to get the best results. See Fig. 80. Here we have a belt

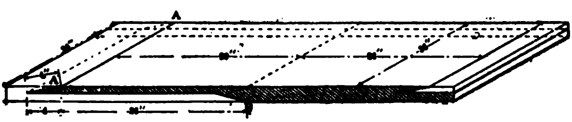


FIG. 80.

that is 36 inches in width and a double ply. Now suppose there is a draft of 9360 pounds on this belt, that from point *A* to point *B* is 26 inches, and that the points are 4 inches long. Now we have 26 inches plus 4 inches plus 4 inches times 36 inches for the number of square inches in the glued joint. This equals 1224 square inches; the total pull on the belt divided by 1224 will equal the load on each square inch of glued joint, and will equal in this case 7.65

pounds. Now instead of assuming distance *A — B* in Fig. 80 to be 26 inches, let the lower joint get out of step with the upper ones, and conditions get vastly different. We will suppose that the dimensions are as given in Fig. 81, as was the case with a new belt that was measured less than one month before the observation was made and we have the following: Joint *A B* is now only 10 inches, and we have 10 inches plus 4 inches plus 4 inches times 36 inches which equals 648 square inches, and the lead on the joint is now 14.44 pounds. You will readily perceive what an important

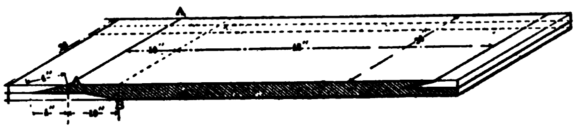


FIG. 81.

part in the life of the belt, and the life of everything around the belt as far as that goes, the proper breaking of the upper and lower joints is. Of course the belt maker will tell you that his glue is just as strong as the leather itself, and he is about right as long as you keep the belt free from oil and water; but when the belt becomes filled with oil the glue rots and loses its strength much faster than does the leather.

No good belt needs any posts along the sides to make it run straight and stay on the pulleys. If the pulleys are in line and the belt straight, it will run straight. All belts should be made to run perfectly straight on pulleys, first on account of the local advertisement that it gives to the man who has charge of them;

second, if they do not run true, they will be on the floor or wrapped around the shaft in a very few minutes, should they ever slip. Another very important thing in the care of belts that carry heavy loads is that if any of the points do come loose so far back that they will not return to place without putting on the clamps, put them on by all means; as the restoring of this point to place means that you will still retain in service all of your belt, as you will not do if you glue it down where it is and thereby cut one side completely out of service.

How to Clean Belting

We submit the following as the best and proper way of cleaning a leather belt. It may seem simple, but it is safe and effective, as has been proved by many people who have thus restored old and dirty belting which had become almost or quite unfit for use.

Coil the belt loosely and place it on edge in a tank in which it may be covered with naphtha; a half barrel makes a good receptacle, but something with a tight cover would save the loss by evaporation. Put in enough naphtha to cover the belt completely and allow it to remain for ten or twelve hours; then turn the belt over, standing it upon the other edge. The vertical position of the belt surfaces allows the dirt to settle to the bottom of the receptacle as it is washed out, and permits naphtha to get at all the parts.

After the belt has remained in the naphtha another ten or twelve hours, or until sufficiently clean, raise it and allow the naphtha to drip back into the tank. Then lay the belt flat, stretching or shaking it until almost

dry. You will find that the naphtha will not affect the leather nor the cement in the center of the belt, but may open the joints at the edges; in which case the old cement should be scraped off and the edges recemented. Your belt man will know how to do this. The belt will now be somewhat hard, and should be treated with a reliable belt dressing before being replaced on the pulleys.

X

BELTING, ITS USE AND ABUSE[1]

THERE is no class of appliances so little understood by the ordinary steam engineer and steam user as belts, which may be seen by the quantity of belting sold annually. Where one can point to a belt that has been in continuous use for twenty years, you can find hundreds that do not last one-fourth as long. Why? Not always because the buyer has tried to get something for nothing, but as a rule, when they do, they get nothing for something.

The average belt is a poor one, and the average buyer will not find it out till he has used it for some time. If you weigh the belt dealer up as a man who is trying to rob you, beat him down in price, then get him to give from 5 to 40 per cent. off, he will enter a protest, and, after some explanation, will come to some terms with you. Have you gained anything by your cleverness? Well, hardly. Belt dealers and makers, like almost all other dealers in supplies, aim to get nothing but first-class goods; but second and third, and even fourth-class goods, are made, and you get the quality you pay for. In the second place, belts wear out quickly because they do not get proper care.

[1] Contributed to Power by Wm. H. McBarnes.

To let a belt run one moment after it gets too slack is bad practice, for it is apt to slip and burn all the staying qualities out of it. Another good reason why it should not be run slack is that the engineer or belt man, to save work, would be tempted to put on a dressing or, worse yet, put on resin to make it pull, and, in the language of Rex, "the man who will put resin on his belts is either a fool or a knave," for it is sure to spoil his belt if continued for any length of time.

In an emergency, as when some unforeseen substance has found its way to the belt, it may be necessary, to keep from shutting down between hours, to use some of the so-called dressing. We know from experience that engineers will go to almost any extreme to get out of a tight place — circumstances sometimes make it necessary to keep a belt running when it should not — but this should not be allowed to any extent. To allow a belt to run too tight is just as bad, for it will make short life for the belt, hot boxes and scored shafting. There is not one in twenty who takes the time or can splice a belt properly; it is generally done in a hurry, any way to make it hold together, with the understanding that it cannot talk; but it does. How often we see boards nailed up or rims tacked on to keep belts from getting off the pulleys. All of this is good for the belt dealers.

This is not all the fault of the engineer or the belt manufacturer. Often belts are made uneven, and soon get out of shape, even with the best of care. We sometimes find a belt that ordinarily runs easy on the pulleys and does its work with ease suddenly inclined to run

to either one side or the other of the driven pulley. This is caused by one of two things — either the belt has been too slack, or the load increased for want of lubrication, or other causes. In either case it will run off if you insist on applying the power. The remedy would be to take up the belt, thoroughly oil the journals, or take off the extra load — maybe a combination of all. Still a little extra work making the belt tighter will enable it to run well and do the extra work just as long as the extra tension can be maintained. Then it may appear perplexing and run to one side of the driven pulley when the driven shaft gets out of line with the driving shaft. In a case of this kind the belt does not run to what is called the high side of the pulley, but to the low side. Another peculiar indication: If two shafts are parallel and there is a high place on the pulley, then a belt will run to the high place; but if the shafts are out of line, or, in other words, are not parallel, and the face of the pulley straight, then the belt will run to the low side or that closest to the driving shaft. The remedy would be to line up your shafting.

The object of this chapter is not to say how belts are made, but to impress upon the minds of belt users that to get the best results, belts, like all good servants, must be well cared for, and all responsibility should rest with one man, just as with your engine or any high-priced machine.

XI

A COMPARATIVE TEST OF FOUR BELT DRESSINGS[1]

DURING January, 1905, a comparative test of the working efficiency of four belt dressings and preservatives was made by T. Farmer, Jr., and the writer. The test was made on the regular belt-testing machine of Sibley College, Cornell University, a full description of which appeared on pages 705-707 of Vol. 12, *Trans. A. S. M. E.* This machine tests the belt under actual running conditions, though our belts were in somewhat better than average condition. The four belts were new 4-inch Alexander No. 1 oak-tanned single-ply, and were 30 feet long. Particular care was taken to keep them free from oil and dirt. The belts were first tested as received from the manufacturer, after which each belt was treated with one of the dressings and again tested.

The dressings were two semi-solids, designated No. 1 and No. 2; a bar, No. 3, and neatsfoot oil, No. 4. As the first three are proprietary articles, it was not thought best to give their names, though any one familiar with the actions of belt dressings will readily recognize No. 1 from its peculiar curve. In applying

[1] Contributed to Power by William Evans.

the dressings, we followed directions carefully, and in the case of Nos. 2 and 3 exceeded them. The belt was given a five-hour run, during which two or three applications of the dressing were given, and then it was set aside in a warm place to allow it to absorb the applied dressing. After thus "soaking" for at least forty-eight hours, the belt was again run, this time for three hours, with one more application of the dressing. As No. 3 was a bar of sticky dressing, it will readily be seen that this precaution was not really necessary. No. 4, the neatsfoot oil, was not applied during the last run, as we were afraid of getting too much oil in the belt. As this oil is so extensively used by engineers for dressing belts, special care was taken to get the best possible results with it.

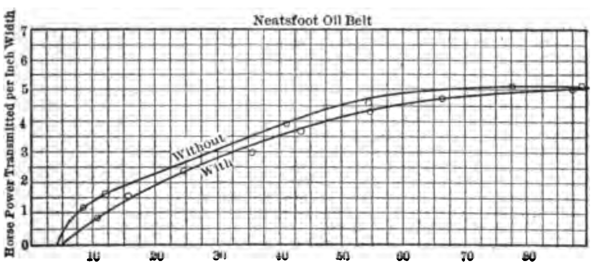


FIG. 82.

In Fig. 82, the result of the test with the neatsfoot oil is shown graphically. This curve is platted to show the relation between initial tension per inch of width and horse-power per inch of width. One reason for the drop in horse-power in the treated belt is that the slip

was materially increased; in the lowest tension at which any power at all was transmitted, about 15 pounds per inch of width, the slip ran up as high as 25 per cent.

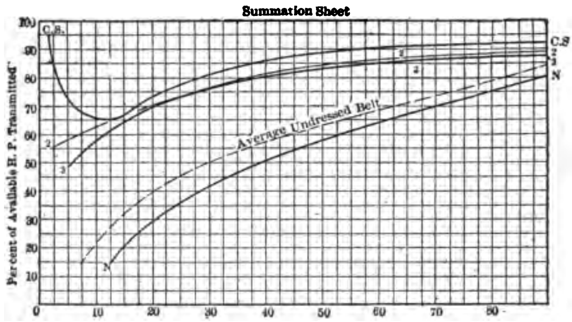


FIG. 83.

In Fig. 83, which shows the comparative value of the four dressings, the highest horse-power delivered to the belt was taken as the standard. The horse-power delivered by the belt was divided by this standard, and the result, expressed in percentage, was used as the percentage of available horse-power transmitted. This comparison shows the great superiority of dressing No. 1 at all times, and especially at low tensions. In looking at this chart, it is well to remember that No. 3 is a sticky dressing.

As the time of the test was so short, we were unable to determine the ultimate effect of the dressings on the leather of the belts. We could only approximate this by a chemical test and a close examination of the belts at the end of each test. The chemical analysis showed

no ammonia or rosin in any of the dressings; No. 2 had a trace of mineral acid, and all had oleic acid as follows: No. 1, 0.27 per cent; No. 2, 29.85 per cent; No. 3, 3.5 per cent; No. 4, 0.7 per cent.

The practical test showed no ill effects except from No. 3, the sticky dressing, which ripped and tore the surface of the belt. The high initial tensions caused overheating of the journals, even though we kept them flooded with oil. On the low initial tensions there was no tendency to heat, even when the maximum horse-power was being transmitted by dressing No. 1. In the latter case we oiled the bearings once in every two or three runs (a "run" comprised all the readings for one initial tension), while in the former we oiled the bearings after each reading and sometimes between them; even then we were afraid that the babbitt would get hot enough to run. The readings for each run varied in number from two to a dozen, but only the one giving the maximum horse-power was used in drawing the curves. The belt speeds during the tests varied between 2000 and 2500 feet per minute, most of the tests being made at about 2200 feet per minute.

XII

BELT CREEP

THE question of the minimum amount of slip of a belt in transmitting power from one pulley to another reduces itself to a qestion of creep, for it is possible to have belts large enough so that with proper tensions there will be no regular slip. With a difference in tension on the two sides and of elasticity in the belt, creep, however, is bound to take place. What does it amount to and what allowance should be made for it? asks Prof. Wm. W. Bird of the Worcester Polytechnic Institute in his paper under the above title.

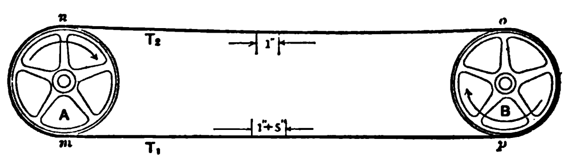


FIG. 84.

In Fig. 84 let A be the driver and B the driven, T_1 the tension in the tight side of the belt and T_2 in the slack side, the pulleys and belt running in the direction indicated. One inch of slack belt goes on to the pulley B at o; at or before the point p it feels the effect of increased tension and stretches to $1 + s$ inches.

It now travels from p to m and goes on to pulley A while stretched. At or before reaching the point n, as the tension decreases, it contracts to one inch and so completes the cycle.

With a light load the belt creeps ahead of the pulley B at or near the point p. If the load is heavy, the creep works towards the point o and the belt may slip; this also takes place when the belt tensions are too light even with small loads.

The point may be easily appreciated by imagining the belt to be of elastic rubber. Professor Bird gives formulas for calculating the creep, and tests made at the Polytechnic to determine the modulus of elasticity. He concludes that the answer to his opening question is that for the common leather belt running under ordinary conditions the creep should not exceed one per cent. While this is sometimes called legitimate slip, it is an actual loss of power and cannot be avoided by belt tighteners or patent pulley coverings.

The smooth or finished side should go next to the pulley because the actual area of contact is greater than when the rough side is in contact; consequently, the adhesion due to friction is greater. Moreover, the smooth side has less tensile strength than the rough side, so that any wear on that side will weaken the belt less than wear on the other side would.

XIII

ROPE DRIVES[1]

THERE seems to be considerable difference in opinion regarding the various ways of applying rope to the sheaves in rope driving, viz., multiple- or separate-rope system, continuous-wrap or single-rope system with the rope from one of the grooves running on a traveling take-up device, continuous-wrap or single-rope system with the take-up working directly on all the wraps.

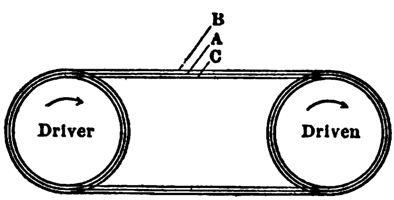


FIG. 85.

The multiple- or separate-rope system on a horizontal drive where the distance between centers is great enough so that the weight of the rope will give the required tension, having the tight or pulling part on the lower side and the sheaves of the same diameter, as in Fig. 85, should be very satisfactory, as old or worn

[1] Contributed to Power by R. Hoyt.

ropes may be replaced by new ones of larger diameter, or some of the ropes may be tighter than others and still not alter the efficiency of the drive. It will be noticed in this case that a larger rope does not alter the proportional pitch diameters of the rope on the driving and driven sheaves; but if one of the sheaves is larger than the other, as in Figs. 86 and 87, and a new or larger

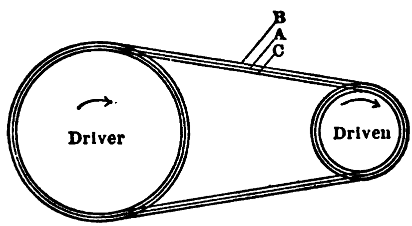


FIG. 86.

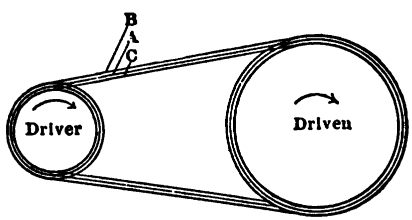


FIG. 87.

rope is substituted for a worn or smaller one, or if some of the ropes are a great deal tighter than others, a differential action will be produced on the ropes owing to the fact that the larger or slack rope will not go as deeply into its grooves as the smaller or tight one.

Consequently the proportionate pitch diameter on the rope on the driver and driven sheave will be changed. The action will depend upon whether the large or the small sheave is the driver. If the driver is the larger, and of course assuming that the slack or large rope is weaker than the combined tight or smaller ones, then it will have less strain on the pulling side; but if the driver is smaller, then the new or large rope will have greater strain on the pulling side. Whether the driver is larger or smaller, a large or slack rope affects the action oppositely to a small or tight rope. Fig. 87 shows how the action is reversed from Fig. 86.

For clearness we will exaggerate the differences in diameter in the sketches and figure the speeds that the different size ropes would produce. We will take *A* as normal, *B* 1 inch farther out of the groove, producing a difference in diameter of 2 inches; *C* 1 inch deeper in the groove, producing a difference in diameter of 2 inches. In Fig. 85 assume for the normal diameter of driver and driven 40 inches, 42 inches for *B* and 38 inches for *C*, with a speed of 200 revolutions per minute for the driver. Either *A*, *B* or *C* will give 200 revolutions per minute for the driven sheave, omitting slippage, of course. In Fig. 86 say the normal diameter of the driver for rope *A* is 60 inches and of the driven 30 inches, a speed of the driver of 200 revolutions per minute will give the driven sheave a speed of 400 revolutions per minute; *B*, with the driver 62 inches and the driven sheave 32 inches diameter, will give the latter a velocity of 387½ revolutions per minute. With *C* the driver is 58 inches, the driven 28 inches, and

the speed given the latter $414\frac{2}{7}$ revolutions per minute. In Fig. 87, the normal diameter of the driving sheave being 30 inches and the driven 60 inches, a speed of the driver of 200 revolutions per minute will give a speed of the driven member of 100 revolutions per minute. With *B*, if the driver is 32 and the driven 62 inches, the driven sheave will have a speed of $103\frac{7}{31}$ revolutions per minute; *C*, with the driver 28 inches and the driven sheave 58 inches, will give the latter a speed of $96\frac{16}{29}$ revolutions per minute. So it will be readily seen what effect a large or a small rope would have.

There are some who claim that slack ropes will transmit more power owing to more wrap on the sheaves, while others claim that tight ropes are better. If a drive with all the ropes slack gave trouble by the ropes slipping, the first remedy tried would be tightening the ropes. But if the conditions were like Fig. 87, it would not be particularly harmful to have some of the ropes

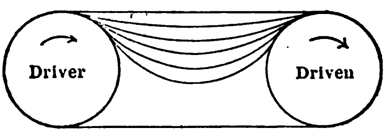


FIG. 88.

longer than others; in fact, it might be well, as the longer ropes would not make a complete circuit as quickly as the shorter ones; consequently the position of the splices would be continually changing. However, it seems more natural to have about the same pull on all the ropes, that is, not have them as shown in Fig. 88.

In conclusion for the system, it should be noted that it has no means of tightening the ropes except by re-splicing; it is not as well adapted to various conditions as the other forms; it is the cheapest form to install and in some cases should give excellent satisfaction.

With the continuous-wrap system having the rope from one of the grooves pass over a traveling take-up, the latter has a tendency to produce an unequal strain in the rope. In taking up, or letting out, the rope must either slide around the grooves, or the strands having the greatest pull will wedge themselves deeper into the grooves, producing a smaller pitch diameter than the ones having less pull, making a differential action on the ropes. It is therefore probable that it is the differential action that takes up or lets out the ropes, the take-up merely acting in a sense as an automatic adjustable idler. In tightening, when the rope stretches or dries out, or even in running normal, the greatest pull will be near the take-up, but if the drive is exposed to moisture, and the rope shortens, it will be farthest from the take-up, depending proportionately on the number of grooves the take-up controls; so in large drives it is best to have more than one take-up.

If one should use an unyieldable substance, as, for experiment, a plain wire on two drums wrapped a number of times around and also over a take-up, and the drums were moved together or apart, he would find that the wire would have to slide around the drum; but, of course, with a rope in a groove it is different. The rope will yield some. It will also go deeper into

the groove. This system costs more than the preceding form, owing to extra expense for the traveling take-up, but may be applied readily to different conditions and will be quite satisfactory in general, if properly designed and installed.

The continuous-wrap system with a take-up or tightener acting directly on all the wraps has practically none of the objectionable features mentioned in the other two forms, and is quick in action, making it applicable where power is suddenly thrown on or off. If the tightener is made automatic, it may be controlled in numerous ways, as with a weight or weight and lever or tackle blocks and weight, etc. It also may be fitted with a cylinder and piston, with a valve to prevent too quick action if power is suddenly thrown off or on. There is ordinarily practically no unequal strain on the rope. This system may be applied to different conditions as readily as the preceding form. Its cost is more than that of either of the others, as the tightener must have as many grooves as there are wraps. It must also have a winder to return the last wrap to the first groove, and to give its highest efficiency it must be properly designed and installed.

In either of the continuous-wrap systems, if a portion of larger rope is used, it will produce a greater strain directly behind the large rope, owing to its traveling around the sheave quicker. In angle work there is always extra wear on the rope in the side of the groove, as only the center or one rope may be accurately lined; so it is not advisable to crowd the centers in angular drives, as the shorter the centers and wider the sheaves

the greater the wearing angle. It must be remembered that the foregoing applies to ordinary simple drives as shown in the sketches; where the drive is complicated, it may be necessary to make other allowances.

XIV

A NEW SCHEME IN ROPE TRANSMISSION[1]

THE use of manila rope for transmitting power is becoming so common as to attract no comment, and it possesses so many advantages in its own field over any other method of conveying power that some objections really existing are overlooked. When a rope drive is installed according to modern practice, it is generally so successful and furnishes such an agreeable and smooth running drive that any possible objection is silenced by the many good qualities it evidently has. But, as a matter of fact, the American continuous method of installing a rope drive has a few serious drawbacks.

Were it possible to install a drive of say thirty ropes in such a manner that each one of the ropes had exactly the same strain on it that each other rope had, and this under varying conditions of speed and load, it is evident that the thirty ropes would work exactly as a belt of proper width to carry the load would, that the ropes would be running with exactly the same tension clear across the width of the drive, like the belt. But according to the best authorities on rope transmission, this ideal condition is impossible to obtain.

[1] Contributed to Power by Geo. F. Willis.

It is given as desirable, by writers on rope transmission problems, to use a take-up sheave for every twelve ropes, while ten is considered even better. The best results have been secured by using a take-up sheave for not more than eight ropes. But in any case the evil of differential driving still exists.

In truth, the only drive in which perfect conditions can exist, according to present practice, is one using but a single rope.

It is evident that when the load comes on the ropes, the entire number of ropes in use are only able to ultimately reach the same tension from the elasticity of the ropes themselves, as slipping in the grooves rarely occurs. But there is a continued and uneven strain on the ropes until the load becomes divided between them, and where ropes are used to drive a varying load, this strain must and does reduce the life of the ropes materially.

Many rope transmissions have been unsatisfactory because of this, and when these drives have been so badly designed as to use one take-up sheave for more than ten ropes, they are apt to be more expensive and troublesome than could have been anticipated.

One rope drive is known where thirty ropes are used, with only one take-up sheave. It has been a source of continual trouble and expense, and has been replaced by the English system of multiple ropes. The inherent troubles of this system have made the changed drive even worse than the original. It will now be replaced by the system here illustrated.

In Fig. 89 is shown a plan view of the tighteners for

a thirty-one rope drive. As the ropes shown are 1½ inches in diameter the main tightener sheave is shown 60 inches in diameter or forty times the diameter of the rope used. Mounted above the thirty-two groove sheave, and in the same frame, is a single groove sheave

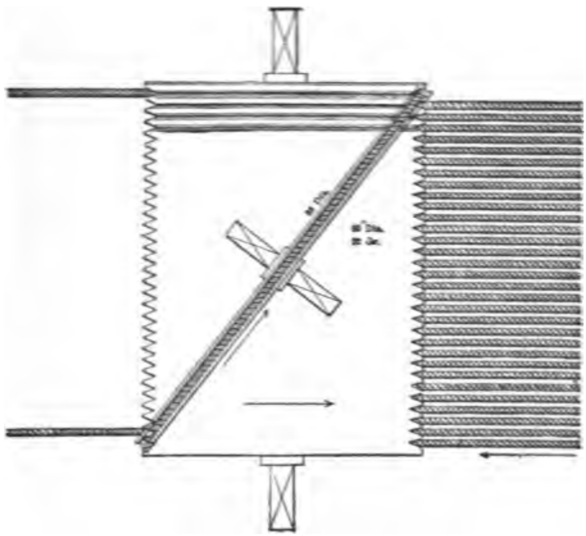


FIG. 89.

of the right diameter to reach the two outside ropes as shown, in this case 86 inches in diameter. Further details are shown in the end elevation, Fig. 90, and in the side elevation, Fig. 91. Allowing a working strain of say 250 pounds to each strand of the thirty-one ropes, we have a total weight of 15,500 pounds which

these two idler sheaves should weigh, including the frame holding them.

These sheaves and the frame are mounted directly upon the ropes, on the slack side of course, and just as a tightener is mounted on a belt. The first rope passes around the thirty-two-groove sheave, on up over the single-groove sheave, and back under the multiple-groove sheave again, and is thus crossed over.

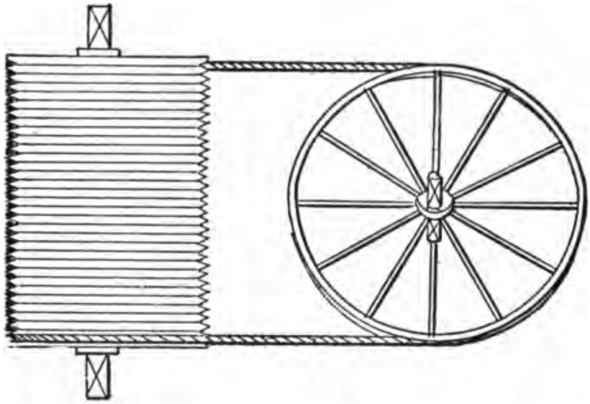


FIG. 90.

It is evident that a rope threaded on this drive would, by the time it had run ten minutes or so, have every strand in exactly the same tension every other strand was in, and that the ropes would remain in this condition in spite of variation of load and speed, as long as they lasted.

The initial expense, including the erection, would probably be no more than that for the necessary six

or eight single-groove idlers, with their shafts and boxes, tracks, etc., which would be necessary according

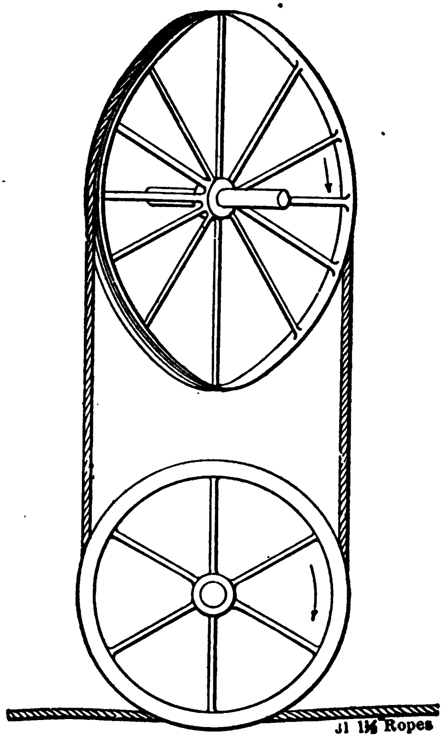


FIG. 91.

to established practice. The room taken up would evidently be much less.

In Fig. 92 an assembled drive of this character is

shown. In Fig. 93 is shown a reverse drive, common in sawmill practice, where the two sheaves described would preferably be mounted on a car, with the proper weight to give the desired tension.

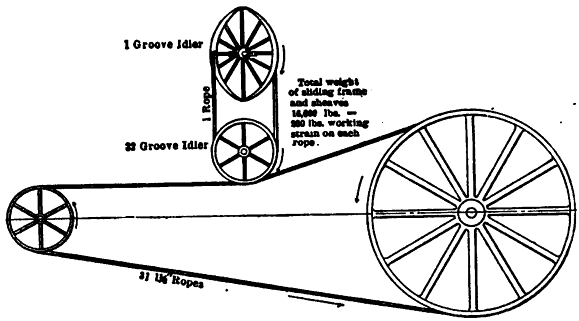


FIG. 92.

In a recent design is shown a cylinder with about 6 feet of piston travel, provided with a reducing valve, so that the steam pressure would remain constant at about 40 pounds. The cylinder is bolted to the mill

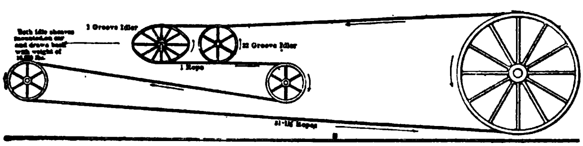


FIG. 93.

frame, while the piston rod is connected to the car carrying the tightener sheaves. The cylinder is of the proper area, when furnished with steam at 40 pounds

pressure, to put the correct strain on the ropes. A small steam trap is part of the equipment. This should give a very elastic tension, and so long as steam pressure was at 40 pounds or over, the tension would remain constant. With 6 feet piston travel, it is evident that 372 feet of stretch could be taken out of the rope, an amount entirely out of the question. A dog, or buffer, can be so located as to prevent excessive back travel of the piston and car when steam pressure is taken off.

It is evident that this method can be applied to a drive using any number of ropes.

XV

HOW TO ORDER TRANSMISSION ROPE[1]

It is probable that more different and erroneous terms are used by purchasing agents and engineers when writing orders for transmission rope than are used to describe any other article needed about a mill. A knowledge of how to order clearly just the kind of rope wanted would prevent delays and expense to many plants. Manufacturers of transmission rope constantly receive orders so peculiar in their wording that they dare not venture an immediate shipment, but must first resort to the mails, telegraph or telephone to find out what is really desired, and, of course, these mistakes, following the law of "the general cussedness of things," usually occur after a breakdown at the very time when every minute's delay means a considerable sum of money lost.

There are in this country two manufacturers of cordage who make a specialty of transmission rope, and the names under which their rope is sold are fairly well known to all users of rope drives. In addition to these two concerns, there are, perhaps, three or four other cordage mills which make this grade of rope to some extent. From this comparatively small source

[1] Contributed to Power by F. S. Greene.

many different brands have sprung which, rechristened, find their way to the market under a variety of names, both poetic and classic. These many names lead to frequent delays in ordering. The man who does the splicing at the mill has, at one time or another, heard of a rope glorying in the possession of some fancy title. It is more than probable that some salesman has told him most wonderful stories of what this particular rope can do; consequently when the time comes for a new rope, the splicer goes to the office and asks that so many feet of such and such a rope be ordered. The purchasing agent makes out the order, using this name, and sends it to the manufacturer, who in all probability has never heard of the rope and knows for a fact that it is not the brand under which any of his fellow manufacturers are selling rope. Before the order can be filled, two or more letters or telegrams must be sent and received.

It frequently occurs that manufacturers receive orders specifying brands which never had existence at all, so far as their knowledge goes. One firm recently found in the same mail requests for "Fern," "Juno," and "Elephant" transmission rope, though no such brands have ever been on the market.

Another familiar mistake is the ordering of a certain color yarn in the rope, as if this decoration possessed some peculiar virtue. These colored yarns are simply a question of dye, and the rope in all probability would be better and stronger were they left out.

Then again, we find peculiar wording as to the lubrication of a rope. Some people insist that the rope

shall be "tallow inlaid"; others call for an "absolutely dry" rope or for a "water-laid" rope. All transmission rope, to be of any service whatsoever, must be lubricated and such a thing as a "dry" transmission rope or a "water-laid" one, whatever that term might mean, would be of but small service to the user. Each manufacturer has his own method or formula for lubricating, and if this be a plumbago or graphite-laid rope, and he is asked for an old-fashioned tallow-laid rope, he cannot fill orders directly from stock.

It is unnecessary to name the number of strands, unless you wish a three- or six-strand rope, for a four-strand transmission rope is always sent, unless otherwise specified. It is also unnecessary to say anything about the core, as the rope is always supplied with one, and generally it is lubricated. Frequently five-strand rope is ordered. This is very confusing, as there is such a thing as a five-strand rope, but it is very rarely made. Ordering a five-strand rope is usually brought about through the error of considering the core as a fifth strand.

It is better, though not necessary, to order by the diameter instead of the circumference, as transmission rope is made and usually sold upon diameter specification.

By far the most frequent specifications received call for "long-fiber, four-strand rope with core," and having done this, the purchaser considers he has named all necessary requirements. At the present price of manila hemp, which varies from 7 cents per pound for the poorer grades to 12½ cents per pound for the best, he

may be quoted for such a rope, with entire honesty, anywhere from 11 to 17 cents per pound. To procure long-fiber manila hemp, and twist it into four strands about a core, does not make a proper transmission rope. As the rope will probably be required to run at a speed of from 3000 to 5000 feet per minute and be subjected to rapid and constant bending throughout its entire length, the fiber should not only be long, but the rope should be soft and pliable. Further than this, as the fiber, yarns and strands must slip one upon another during the bending, the rope should be so lubricated as to reduce to a minimum the frictional wear from such slipping and rubbing, which is a much larger factor than is generally supposed. Again, the unusual strength of manila fiber is shown only when subjected to a longitudinal strain. Transversely, owing to the cellular formation, the fiber is relatively weak; therefore, in manufacturing transmission rope, the greatest care is necessary to secure such proportion of twist in both yarns and strand as to render the rope least vulnerable to crosswise strain. Nor will the term "long fiber" insure the purchaser obtaining the proper material in his rope, for the longest manila fiber, contrary to general belief, is not always the best from which to make a transmission rope. Some of the extremely long variety is coarse and brittle. The best fiber for transmission rope is a particular grade of manila hemp known as Zebu, Fig. 94, which is light in color, silky to the touch and exceedingly strong and flexible.

The accompanying illustration, Fig. 95, shows a close view of two grades of hemp, that on the left being

FIG. 94.

FIG. 95.

known in the trade as "Superior 2ds," while the fiber to the right of the cut is "Zebu." Fig. 96 shows

FIG. 96.

a more distant view of the same two "heads" of hemp, and the reader will see that in both the fiber is exceed-

ing long, and if anything, that of the Superior 2ds is longer than in the Zebu. A transmission rope made from the latter, however, will cost the manufacturer from 3½ to 4 cents more per pound than if he had used Superior 2ds, and will outlast two ropes made from the longer though coarser fiber.

The reader, if he has perused this chapter to the present point, is doubtless now asking himself: "How shall I word my order when I want a first-class driving rope?" The safest road to follow is to write to some manufacturer or firm whom you know to be reliable, and ask for so many feet of their transmission rope, giving the name, if you are certain on that point, and, of course, being sure to mention the diameter. In case you do not know the name of his rope, word your order as simply and briefly as possible; for example: "One thousand feet 1½ inches diameter first quality manila transmission rope," and if the concern to which you write is a reputable one, you will receive a four-strand rope, made from Zebu manila hemp, put together with proper twist and lay for the service required.

XVI

A BELTING AND PULLEY CHART[1]

RULE 1. *Pulley Speed.* — When the diameter of both pulleys and the speed of one is given, to find the speed of the other: Place the points of spacing dividers upon the two given diameters in inches upon the scale (Fig. 97); then raise the dividers, keeping the space obtained, and place one point on the given speed and the other *above* it for speed of *S*, or *below* it for speed of *L* (*S* and *L* meaning smaller and larger pulley, respectively). This point will fall upon the required speed.

Example: If the two pulley diameters are 10 and 25 inches and speed of larger pulley is 120 revolutions per minute, what is speed of small pulley?

Place the points of dividers on 10 and 25 on scale *A*, then lift the dividers and place one point on 120 and the other above it upon the scale; the other point now rests on 300 as the speed of *S*. If the speed of *S* had been given, one point would have been placed at 300 and the other *below* it, falling upon 120, the required speed of *L*.

Note. — In applying this rule, if the speed comes beyond the range of scale *A*, the result may be read

[1] Contributed to Power by A. G. Holman, M. E.

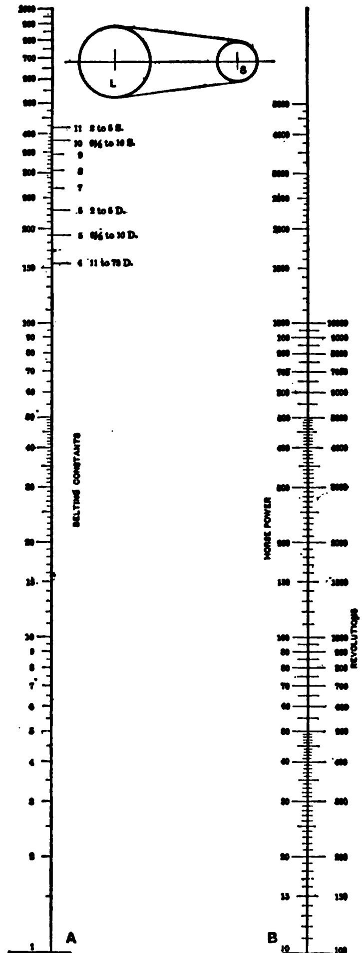


FIG. 97.

by carrying the space to the revolution scale on scale *B*, and proceeding in the same way.

Example: Diameter of pulleys 12 and 36 inches and speed of *L* 500, what is speed of *S?* Place points of dividers on 12 and 36. Now, if dividers are raised and one point placed on 500 and the other above it on scale *A*, it will come beyond the top of the scale. Hence go to scale *B*, placing lower point on revolution scale at 500 and the other point above, which will fall upon 1500, the answer.

RULE 2. *Pulley Diameters.* — When the speed of both pulleys and the diameter of one is given, to find diameter of the other: Place points of dividers on the two speeds on scale *A* or revolution scale *B*. Then place one point of dividers on given diameter and the other above it to find diameter of *L*, or below it for diameter of *S*. The figure thus indicated is the required diameter.

Example: Speeds 180 and 450 and diameter of smaller pulley 20. What must be diameter of *L?*

Place points of dividers on 180 and 450 on scale *A*. Then place one point on 20 (the given diameter). The other point falls at 50, the required diameter of *L*.

If the point falls between two graduations in any problem, the result can be closely judged by the relative position.

The other and more labor-saving use for this chart is its application to belting problems. It is generally conceded that there is no subject of more general interest in practical mechanics and none on which there is a greater difference of opinion than the proper allowance

to be made in the selection of belt sizes for given requirements. The general formula for the horse-power transmitted by belting is

$$HP = \frac{WS}{C} \text{ in which } HP = \text{horse-power,}$$

W = width of belt in inches, S = speed of belt in feet per minute, and C = constant.

The proper values of this constant, or the feet per minute that each inch of width must run to transmit a horse-power, under certain conditions, is the point in question.

On the right-hand side of line A on the chart is a series of lines representing different values for this constant. The lower one, marked 4, represents 400 feet belt speed per minute, the next above is for 500, and so on. Against some of these values are suggestions as to belts often recommended in connection with these constants. For instance, 2 to 6 S suggests the constant 1100 to be used for 2- to 6-inch single leather belt, 1000 for 6½- to 10-inch single, 600 for 2- to 6-inch double, etc.

These suggestions practically agree with the advice of the Geo. V. Cresson Company's catalog and the deductions of Kent's Handbook.

More power may be transmitted than these suggestions will allow, by increasing the tension, but this is accompanied by the disadvantage of requiring extra attention and undue pressure upon bearings.

The use of the chart for horse-power and width of belting is explained by the following rules:

Rule 3. *Horse-power of Belting.* — To find the horse-power that can be transmitted when diameter and speed of pulley and width of belt are given: Place one point of dividers on scale *A* at the width of belt in inches and the other point at the bottom of the line (at 1). Next add this space to the hight representing diameter of pulley by placing lower point of dividers upon the given diameter and allowing the other point to rest upon the scale above. Then holding the upper point stationary, open or close dividers until the other point falls upon the proper constant on the scale at right-hand side of line *A*. Now transfer this space last obtained to the scale *B* by raising the dividers, carrying them square across to *B* and placing the point that was on the constant upon the given speed on the revolution scale. Note the location of the other point of dividers upon the horse-power scale, which indicates the horse-power that can be transmitted under the given conditions.

Example: What horse-power can be transmitted by an 8-inch double belt running on a 40-inch pulley at 500 feet per minute? Place one point of dividers on line *A* at 8 (width of belt) and the other point at bottom of line. Next raise dividers and place lower point on 40 (diameter of pulley) and let the other point fall above upon the scale. Then close dividers until lower point comes to the constant for 6½ to 10 double. Carry this space to scale *B* with lower point on 500 on revolution scale. Under point now falls upon 84 on horse-power scale, which is the required horse-power.

Rule 4. *Width of Belting.* — To find the necessary

width of belting when size and speed of pulley and the horse-power are given: Place one point of dividers on scale *B* upon the horse-power and the other point upon the revolutions. Next transfer this space to scale *A* by raising the dividers, carrying them square across and placing the point that was on revolutions upon the constant. Then holding the other point stationary, raise the point that was on the constant and open dividers until this point falls upon the given diameter. Now lift the dividers and carry the lower point down to bottom of line (the point 1). The upper point will now indicate the required width of belt.

Note. — If, in finding width of belt, there is doubt about the proper constant to take, a medium value, say 6, may be assumed and a hasty "cut and try" will show in what classification the required belt will come.

Example: What width of belt for 100 horse-power with 40-inch pulley at 500 revolutions?

Place point of dividers on scale *B* upon 100 on horse-power scale and the other upon 500 on the revolution scale. Then carry the space to scale *A* with lower point on constant 5. Then resting dividers upon upper point open them until lower point is at 40 (diameter). Finally, raise dividers and place lower point at bottom of line. Upper point is now at 9½, indicating the nearest even width 10 as the answer.

A little practice will make one familiar with these rules, and it will be seen that in the belting rules the four motions perform two multiplications and a division.

XVII

SPLICING ROPE

THE splicing of a transmission rope is an important matter; the points on which the success of the splice, and incidentally the drive, depend being the length of the splice, which in turn depends upon the diameter of the rope and which is given in the table (Fig. 97a);

DATA RELATIVE TO MANILA TRANSMISSION ROPE AND SHEAVES

Diameter of Rope in Inches	Square of Diameter	Approximate Weight per Foot, Pounds	Breaking Strength, Pounds	Maximum Allowable Tension, Pounds	Length of Splice in Feet			Smallest Diameter of Sheaves in Inches	Maximum Number of Revolutions per Minute
					3-Strand	4-Strand	6-Strand		
½	.25	.12	1750	50	6			20	1060
⅝	.2906	.16	2730	80	6			24	970
¾	.5625	.20	3950	112	6	8		27	760
⅞	.7656	.26	5400	153	6	8		32	650
1	1.	.34	7000	200	7	10	14	36	570
1⅛	1.2656	.43	8900	253	7	10	16	40	510
1¼	1.5625	.63	10,900	312	7	10	16	45	460
1½	2.25	.77	15,700	450	8	12	18	54	380
1¾	3.0625	1.04	21,400	612	8	12	18	63	330
2	4.	1.36	28,000	800	9	14	20	72	290
2¼	5.0625	1.73	35,400	1012	9	14	20	81	255
2½	6.25	2.13	43,700	1250	10	16	22	90	230

FIG. 97a.

the diameter of the splice, which should be the same as the diameter of the rope; the securing of the ends of the strands of the splice, which must be so fastened that they will not wear or whip out or cause the overlying strands to wear unduly; and the workmanship of the splice, which should be the best it is possible to secure. When splicing an old and a new piece of rope, the new piece should be thoroughly stretched, for, at best, it is an exceedingly difficult task on account of the stretch and difference in diameter of the rope.

The illustrations and instructions for making standard rope splices are taken, by the courtesy of the American Manufacturing Company, from their "Blue Book of Rope Transmission."

There are many different splices now in use, but the one that experience has proved best is what is known as the English transmission splice. In describing this we take for our example a four-strand rope, $1\frac{3}{4}$ inches in diameter, as spliced on sheaves in the multiple system. The rope is first placed around sheaves, and, with a tackle, stretched and hauled taut; the ends should pass each other from six to seven feet, the passing point being marked with twine on each rope. The rope is then slipped from the sheaves and allowed to rest on shafts, to give sufficient slack for making the splice.

Unlay the strands in pairs as far back as the twines *M, M′*, crotch the four pairs of strands thus opened (Fig. 98), cores having been drawn out together on the upper side. Then, having removed marking

twine *M*, unlay the two strands 6 and 8, still in pairs, back a distance of two feet, to *A*; the strands 1 and 3,

FIG. 98.

also in pairs, being carefully laid in their place. Next unlay the strands 5 and 7 in pairs, to *A′*, replacing them as before with 2 and 4. The rope is now as

shown in Fig. 99. The pair of strands 6 and 8 are now separated, and 8 unlaid four feet back to *B*, a distance of six feet from center, strand 6 being left at

Fig. 99.

A. The pair of strands 1 and 3 having been separated, 3 is left at *A*, as companion for 6, strand 1 being carefully laid in place of strand 8 until they meet at point

B. The two pairs of strands 2–4 and 5–7 are now separated and laid in the same manner, every care being taken, while thus putting the rope together, that original twist and lay of strand is maintained. The protruding cores are now cut off so that the ends, when pushed back in rope, butt together.

The rope now appears as shown in Fig. 100, and after the eight strands have been cut to convenient working lengths (about two feet), the companion strands are ready to be fastened together and "tucked"; this operation is described for strands 2 and 7, the method being identical for the other three pairs. Unlay 2 and 7 for about twelve to fourteen inches, divide each strand in half by removing its cover yarns (see Fig. 101), whip with twine the ends of interior yarns 2′ and 7′; then, leaving cover 2, relay 2′ until near 7 and 7′, here join with simple knot 2′ and 7′, Fig. 102. Divide cover yarns 7, and pass 2′ through them, continuing on through the rope *under* the two adjacent strands, avoiding the core, thus locking 2′, Fig. 103. *In no event pass 2′ over these or any other strands.* Half-strand 7′ must now be taken care of; at the right of the knot made with 2′ and 7′, 2′ is slightly raised with a marlin spike, and 7′ passed or tucked around it two or three times, these two half-strands forming in this way a whole strand. Half-strand 7′ is tucked until cover 2 is reached, whose yarns are divided and 7′ passed through them and drawn under the two adjacent strands, forming again the lock. The strand ends at both locks are now cut off, leaving about two inches, so that the

FIG. 100.

FIG. 101.

FIG. 102.

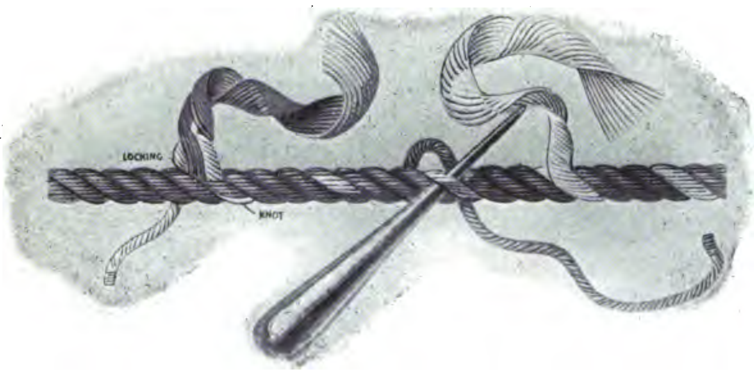


FIG. 103.

yarns may draw slightly without unlocking. This completes the joining of one pair of strands, Fig. 104. The three remaining pairs of strands are joined in the same manner.

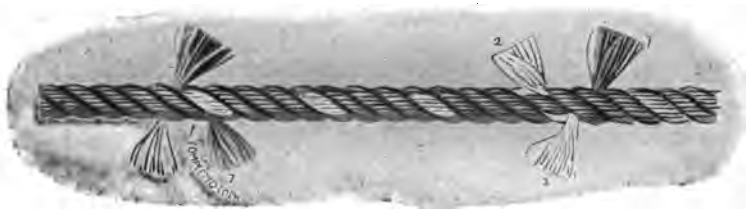


FIG. 104.

After the rope has been in service a few days, the projecting ends at locks wear away, and if tucks have been carefully made, and the original twist of yarns preserved, the diameter of the rope will not be increased, nor can the splice be located when the rope is in motion.

XVIII

WIRE ROPE TRANSMISSION[1]

WIRE ropes are extensively and successfully used in the horizontal and inclined transmission of great power of unlimited amount, the advantages over hemp rope belting being: driving at very long distances, comparatively small loss through slipping and the possibility of driving in the open air.

Vertical transmission of power, on account of the weight of the rope, is excluded.

Formerly the material used in the manufacture of the wires was best charcoal iron, but now almost exclusively tough crucible-steel wires are used, as steel wire ropes are stronger, do not stretch as much, and last longer than iron ropes.

The wire ropes consist of six strands of from six to twenty wires each, and the strands to form the rope are woven in the opposite direction to the wires in the strand. In the center of each strand and in the center of the rope a cotton core is placed. These cores are of the greatest importance, for by reducing the friction of the wires against each other, they serve to increase the lifetime of the rope, which, according to the strain

[1] Contributed to Power by C. Boysen, M. E.

on the rope and the size of the smallest pulley, is from one to three years.

To prevent rusting, the wire ropes receive a coat of boiled linseed oil, or a hot mixture consisting of three parts of drip oil and one part of resin is applied. This latter mixture at the same time improves the adhesion between the rope and the lining placed in the bottom of the pulleys, thus reducing the loss caused by slipping of the rope. Wire ropes used for the transmission of power should never be galvanized.

The ends of the rope are spliced together, from 10 to 20 feet being necessary for a good splice; great care should be taken that the splice is made by experienced men, and that the rope is made long enough. A rope stretches constantly from the time when placed on the pulleys, the more so when placed on the pulleys tightly. Therefore it has to be made long enough to transmit power without undue tension, and for this reason the distance between the two pulleys has to be long enough and the working strain per square inch of section low enough to allow sufficient deflection in the rope. As a guidance to the amount of deflection necessary, be it said that even in a short drive the deflection of the rope, when not running, should not be less than 2 feet; and for a distance of 400 feet between pulley centers, the deflection of the rope when running should be 5 feet in the driving rope and 10 feet in the driven rope.

Either the top or the bottom rope may be the driving one, the former being preferable; but the ropes should never be crossed.

Power can be transmited to a distance of 6000 feet

and more without great loss; but as two pulleys should on no account be more than 500 feet apart, intermediate stations are placed along the road.

Precautions should be taken against the possibility of the rope swaying. This may be caused either by the influence of the wind, by a bad splice, by the rope wearing too much, by the pulleys not being balanced well or by the pulleys not being in the same plane. It is of importance that the pulleys be exactly in line, and careful attention should be given to the construction and placing of the bearings. Although the bearings are not strained excessively, the steps are usually made long and movable. The connection between the shaft and the pulley is best made by means of tangential keys.

Some engineers, when two ropes are found necessary for the transmission of the power in question, use pulleys containing two grooves each, and make the same kind of pulleys for the intermediate stations of long-distance driving; whereas others advise a separate pulley for each rope, both being connected with each other by a clutch.

The diameter of the smallest pulley has to be large enough in comparison with the diameter of the rope or the thickness of the single wires used to easily overcome the stiffness in the rope. The larger the pulleys, the longer the rope will last.

The rim of the pulley is V-shaped, and the bottom of the groove is dovetailed to receive a lining of wood, rubber or leather, on which the rope rests. The lining increases the friction and reduces the loss caused by slipping of

the rope. Leather is the best lining and lasts about three years. Either old belt leather, well saturated with oil, or new leather, boiled in fish oil, can be taken. It is cut in pieces of the same size as the dovetailed part of the groove, and then placed on and pressed together in the latter. The pressing is done by means of a piece of wood. The last remaining small space in the groove is filled with soft rubber. If the lining has to consist of rubber, this is softened and hammered into the groove. For wood lining, thin blocks of the required size are placed into the groove through a hole provided in the bottom of the rim. This slot is closed by a plate and fastened to the bottom of the rim by means of screws after all blocks have been inserted. The lining has to be turned absolutely true, for which reason the filling is done while the pulley is still in the lathe.

Pulleys up to 3 feet in diameter are built with cast-iron arms; whereas larger pulleys have wrought-iron arms made of round iron, cast in the rim and boss. Pulleys under 8 feet 6 inches in diameter are made in one piece, if for other reasons it is not necessary to have them in halves.

Guide pulleys are used for long ropes, especially if there is not sufficient hight above the ground. The guide pulleys are of the same construction as the main pulleys, and for the driving rope they are also made of the same diameter. The diameter of the guide pulleys for the driven rope can be made from 20 to 25 per cent. smaller.

The breaking strength of unannealed wires per square

inch of section and according to thickness and quality is: For iron wires from 70,000 to 110,000 pounds, and for steel wires from 110,000 to 130,000 pounds. For thinner wires a higher value is taken than for thick ones.

The diameter of the wires used for making ropes for transmitting power is from 0.02 to 0.1 inch, and on account of the stiffness, no wires above the latter size should be used. A rope consisting of a greater number of thin wires, besides being stronger is more pliable and lasts longer than a rope of the same area consisting of a less number of thicker wires.

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T

SHAFT GOVERNORS

Hill Publishing Company, New York, U.S.A.

CONTENTS

INTRODUCTION

THIS book is made up from material originally published in *Power*, together with some special articles which have been prepared to make it a complete handbook of the subject. The fact that nowhere in a single book can all of this material be found in a form which will be useful to the practical engineer, will, it is hoped, make the book of special interest and value.

The compiler wishes to acknowledge his indebtedness to a number of men who have contributed brief articles to *Power* and furnished him with special information regarding the various types of governors.

HUBERT E. COLLINS.

NEW YORK, September, 1908.

SHAFT GOVERNORS

I

EVOLUTION OF THE SHAFT GOVERNOR.*

THE development of the shaft governor has been a slow and steady one in this country, commencing probably in 1829, or possibly even later. It is quite probable that for a long time this governor met with little or no practical application, as it is a fact which will appear later that the period of its practical application can hardly be said to have begun before 1876. Since that time the growth in use of this governor in this country has been remarkable and many forms have been produced, all of which possess more or less merit. In England this governor seems to be scarcely known to-day, judging at least from the literature on the subject, while on the continent of Europe its use is also very limited.

My sources of information regarding the development of the shaft governor are principally to be found in the literature relating to the steam engine, which has been published from time to time during the last thirty or forty years, and in the records of the United States patent office.

The general works relating to the steam engine,

* Paper read at the meeting of the Engine Builders' Association, New York, December, 1901, by R. C. Carpenter.

with the exception of a few American works in late years, contain very little in relation to the shaft governor. So far as I can ascertain, all the works published by English authors, even up to a very late date, are entirely silent on this subject; thus, for instance, the work on the steam engine by Prof. John Perry, written in 1899, while devoting a full chapter to the subject of the fly-wheel and governor, and while describing in full the theory and various forms of the pendulum governor, is absolutely silent regarding the shaft governor. So far as I can learn from the literature which has been printed in England regarding the steam engine, any student obtaining his information from such books would know nothing whatever of the structure of the shaft governor.

The French writers on the subject of the steam engine do give considerable information relating to the subject of the shaft governor; the governor is, however, invariably described as an American invention which is used on certain American engines, and one obtains the idea from such a description that the governor is little used in France.

American books relating to the structure of the steam engine published twenty-five years ago entirely neglect the existence of such a governing device, and it seems quite probable that although the shaft governor was used twenty-five years ago to a very limited extent, it had not, at that time, made a sufficiently strong impression on writers as to lead them to consider that it was a practical device. As illustrations of this kind, we note a few instances. Thus, Knight's

Mechanical Dictionary, published in 1877, is a work devoted to explaining the structure of various machines and prime movers, and has never been surpassed or even equaled in its particular field. This work describes in detail the structure of a large number of governing devices and presents a full-page illustration showing the forms of governors supposed to be of practical value. (Fig. 1.) You will notice that some twenty-three different forms are shown, all, however, of the type known as the rotating or swinging pendulum governors, and none belong to the class which it is the object of my paper to describe. In Appleton's Encyclopædia of Applied Mechanics, published in 1878, and edited by the ablest corps of specialists ever employed at that date in this country, is a very full and complete article on the steam engine, but it makes no reference whatever to the use of the shaft governor, which was perhaps inexcusable at that date, as a shaft governor was exhibited at the Centennial Exposition in 1876.

The oldest book which I have in my library containing references to the shaft governor is "Steam Using; or, Steam Engine Practice," written by Prof. Charles A. Smith, of St. Louis, in 1885. In this work are published detailed drawings of a Westinghouse engine, and also a Buckeye engine, and each is shown with a shaft governor. I have no information at hand which enables me to state the earliest dates at which these companies commenced the building of shaft governors on a commercial scale, nor am I certain but that other engine companies introduced the

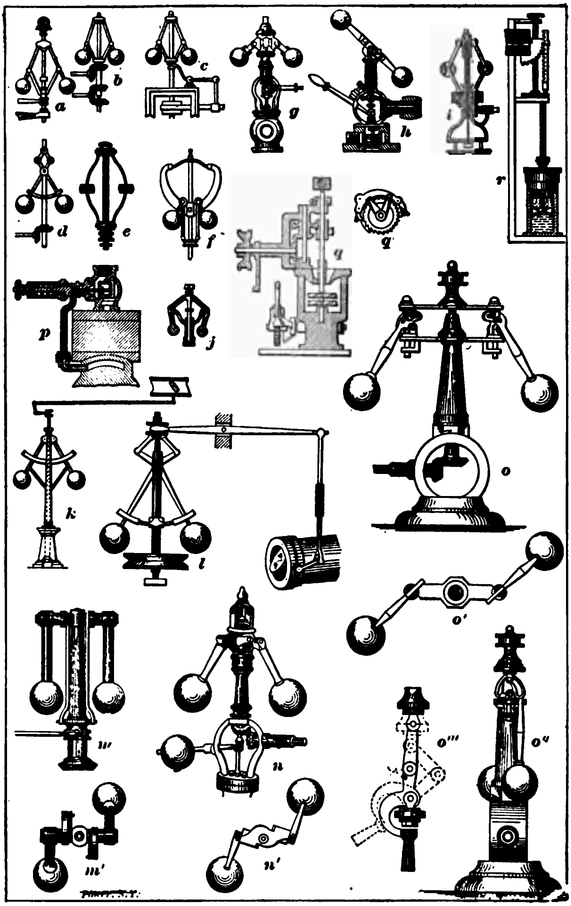

FIG. I

use of the governor at a somewhat earlier date. Views of these governors as given in Professor Smith's work are shown in Figs. 2 and 3. At the Centennial Exposition at Philadelphia, held in 1876, Prof. John E. Sweet showed an engine fitted with a shaft governor which had been built under his supervision by stu-

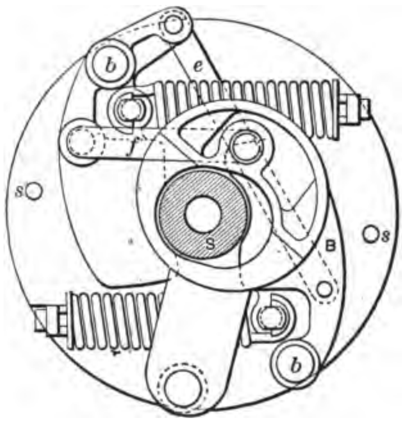

FIG. 2

dents in the shops of Cornell University. This exhibition seems to have been the inspiration which resulted in the construction of the shaft governor by many manufacturers, and the governor shown (Fig. 4) was the pioneer in the later period of development of this important invention.

This was not the first engine constructed by Professor Sweet, but was, I believe, engine No. 3. This Centennial engine is still preserved in the Museum of Sibley College, although the original governor was long ago

removed. The original governor was temporarily removed in 1889 to carry on some experimental work with governors of a different design on the same engine. Some of the parts of the governor were broken and it has never been possible to restore them in the original condition. The shaft governor on the Centennial engine was very different in construction from the later

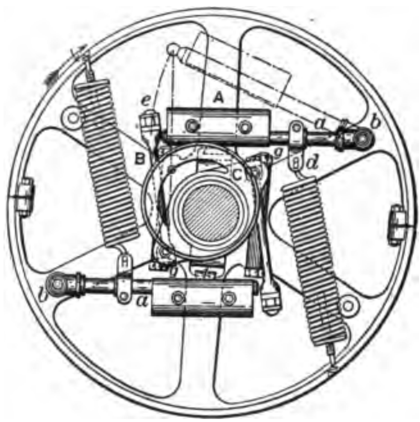

FIG. 3

ones designed by Professor Sweet and from the one now used on the Straight Line engine. The valve-rod was connected to an eccentric through the medium of a geared disk.

In later constructions of the governor applied to the Straight Line engine, the valve is connected to a swinging eccentric by link motions.

My study of the literature of the subject would in-

dicate that the shaft governor is at least, so far as its practical application is concerned, strictly an American invention, and furthermore, this invention has not been introduced to any great extent, even at the present time in Europe, while in England its use is so limited that English writers of text-books have not considered it of sufficient importance to merit any

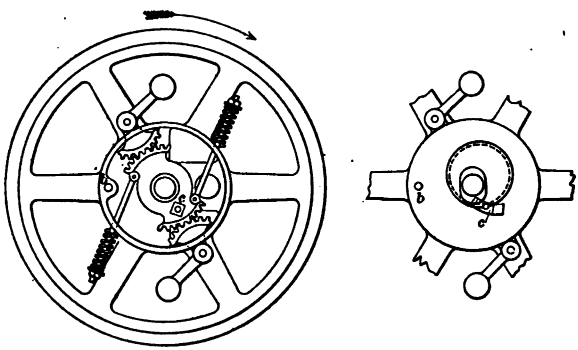

FIG. 4

mention. In this country the steam engine governor has followed the course of every great invention in its development; it has been developed, not by a single person or as a single invention, but rather by the slow and tedious process of experiment and practice. As in the steam engine itself, we find, doubtless, first a period of speculation, during which time theoretical investigations were made and patents taken out, and this period probably extended until about 1870; then comes a period of application, beginning in a small

way perhaps with 1870 and extending through the next fifteen years, during which time numerous applications of various forms were made, tried with greater or less success, modified and improved until finally a high degree of perfection has been reached.

The earlier form of governor and the one which is almost exclusively used in England and other European countries to-day was invented by James Watt, or at least adapted for use on the steam engine by Watt.

It is hardly probable that Watt ever considered himself as the inventor of the governor for regulating the speed of an engine, for the reason that I do not find this invention claimed in any of his patents and, judging from the character of the claims made in his numerous patents, Watt was not the kind of a man to omit protecting himself for any of his inventions.

In the life of Watt, by Muirhead, it is stated that for the purpose of regulating the speed of the engine Mr. Watt tried various methods, but at last fixed upon what he called the "governor," consisting of a perpendicular axis turned by the engine; to a joint near the top of this axis is suspended two iron rods carrying heavy balls of metal at their lower ends, in the nature of pendulums. When this axis is put in motion by the engine the balls recede from the perpendicular by the centrifugal force, and, by means of a combination of levers fixed on their upper end, raise the end of a lever which acts upon the spanner of the throttle-valve and shuts it more or less according to the speed of the engine, so that as the velocity augments the valve is shut, until the speed of the engine and the opening

of the valve come to a maximum and balance each other. The application of the centrifugal principle was not a new invention, but had been applied by others to the regulation of water and windmills and other things; but Mr. Watt improved the mechanism by which it acted upon the machines and adapted it to his engines.

Such, says M. Arago in describing Mr. Watt's application to the steam engine of the governor or regulator by centrifugal force, was its efficacy, that there was to be seen at Manchester a few years ago, in the cotton mill of Mr. Lee, a man of great mechanical talents, a clock which was set in motion by the steam engine used in the work, and which marked time very well, even beside a common pendulum clock.

The principle of action of the simple governor of the revolving pendulum type can be expressed by an equation as follows:

$$h = \frac{gr^2}{v^2} = \frac{g}{4n^2\pi^2}$$

from which

$$n^2 = \frac{g}{4\pi^2 h}$$

$$n = \frac{1}{2\pi}\sqrt{\frac{g}{h}} = \frac{constant}{h}$$

In this equation n equals the number of turns per second, v the velocity in feet per second, r the horizontal projection of the arm of the pendulum, h the vertical projection of the arm of the pendulum, g the force of gravity. These equations are well known and the ex-

planation of their derivation can be found in any treatise on the subject. It is noted that the position of the governor balls which are determined by the quantity h does not vary with the speed of the engine which is represented by the smybol n, but varies with the square of the speed of n^2, consequently a governor of the simple pendulum type cannot be made so as to give a perfectly uniform motion without some change in form or construction not known to Watt. To make the revolving pendulum isochronous in its action many devices have been brought out, and while these have in a great measure improved its action, none of them have been entirely successful. The pendulum governor has been much improved by arranging it to lift a weight and also by crossing the arms of the pendulum and arranging their point of suspension to one side of the axis. By these arrangements the distance passed through by the moving parts of the governor becomes very nearly proportional to the change in motion of the engine. These governors have also been constructed so as to utilize the force of springs instead of that of gravity to counteract the effect of the centrifugal force.

The revolving pendulum governor has usually been constructed to regulate the speed by being attached to a throttle-valve in the steam-pipe, which was opened or closed as desired. It has, however, been employed in a few cases to regulate the motion of the engine by changing the travel of the steam-valve through the medium of a link motion, and in the drop cut-off class of engines to regulate the speed by unlocking the valve

mechanism so as to permit closing, as in the Corliss type of engine.

Where the regulation is accomplished by throttling the steam supply, poor results are generally obtained for reasons entirely independent of the action of the governor, since necessarily more or less time must elapse before the proper amount of steam to give the desired speed can be made to pass through a throttled orifice. The throttling governor as usually constructed in this country has not been of the highest type of workmanship, nor has it accomplished all of the results in regulation which would have been possible with governors of its type and class, made with better design and workmanship.

The formula to which reference has already been made does not consider the retarding effect of friction. There is perhaps nothing so important in its effect on results of regulation as friction, which always acts to resist any moving force; it tends to prevent the governor balls from moving to their true position whether the motion of the engine is too fast or too slow, and consequently it becomes responsible for irregular action of the governor and for much of the imperfect regulation. It is, however, important to note that the revolving pendulum governor is not theoretically perfect, and aside from imperfections of construction and design it cannot be made to give a perfectly uniform motion to the engine.

In the shaft governor we find in every case a weight supported by an arm or arranged to move in guides connected to a revolving fly-wheel, so that the centrif-

ugal force tends to throw it away from the center. A spring is employed to counteract the effect of centrifugal force and is so arranged as to restore the weights to the normal position when the engine comes to rest. In this governor the centrifugal force tends to throw the weighted portions outward and toward the circumference of the revolving wheel, whereas the spring tends to draw the weight inward and counteracts the centrifugal force, holding the governor in such position as to maintain uniform speed. By properly proportioning and arranging the weights and the spring, it is entirely possible to make a governor of this class so that its parts will move directly proportional to any change of speed of the engine, and consequently it will take such a position as will tend to keep the motion perfectly uniform regardless of other conditions. In other words, it is possible to make a governor of this class which will give theoretically uniform motion.

The tendency of a moving body to continue its motion uniformly has been well known since the time of Sir Isaac Newton and is generally known as the "principle of inertia." It has been recognized from the earliest times in the art of steam engine building that heavy fly-wheels conduced to uniformity of motion because of the inertia of the parts. This uniformity of motion is a well-known function of the weight of the fly-wheel. Consequently it has been the practice for years to use heavy fly-wheels where a uniform motion is desired, and even at the present time we have found no system of regulation which entirely permits us to do away with that produced by the inertia

of heavy weights. The irregular motion produced by the intermittent action of the steam on the piston can be very largely reduced to a uniform action by the use of an extremely heavy fly-wheel and the minute variations in speed can probably be controlled by no other method. As the engine is made to revolve at a higher speed the impulses are made at greater rapidity and consequently a fly-wheel of smaller weight can be employed for the same degree of uniformity of motion. The shaft governor could, of course, be connected to a throttle of a steam engine and would in that case produce results superior to any of the revolving pendulum governors, but such an application has, so far, as I know, never been attempted. The governor has been universally connected through the medium of rods and links directly to the main or auxiliary valve which regulates the supply of steam to the engine. The advantage gained by this construction is that of admitting steam of full power behind the piston at each stroke, and thus giving the full benefits of expansion of the steam in its work.

This advantage is great and will result under usual conditions in a marked improvement in economy, as compared with a throttling engine otherwise the same. I had an opportunity once of testing two engines, one automatic, the other throttling, both in excellent condition, doing alternately the same work. The results, which I do not have here in full, showed slightly over 12 per cent. in favor of the economy of the automatic engines, yet the conditions I considered as favorable as possible for the throttling construction.

The shaft governor has proved itself to be especially adapted for engines moving at a comparatively high speed of rotation. The results produced in the way of regulation in engines of this type have been in some instances simply remarkable, as it has been found entirely possible to produce a governor which would hold the engine to the same number of revolutions per minute, whether the engine were running light or loaded, or whether the load were suddenly or slowly applied or removed.

The shaft governor, revolving as it does with the shaft of the engine, is affected by the inertia of its particles in the same manner as the revolving fly-wheel. The governor parts may be arranged so that this inertia effect may tend to make its action quicker, in which case the regulation of the engine would be improved, or it may be arranged so as to have the reverse effect, in which case the regulation of the engine would be worse than before. This effect of inertia on the part of the governor and its use for improving the regulation was not recognized until the shaft governor had been pretty well developed, but a study of the drawings of some of the early types of governors show that they were constructed and operated in such manner as to have the full benefit of inertia to aid in the regulation. This seems to have been notably true in the case of the governor shown by Professor Sweet at the Centennial Exposition.

The records of the American Patent Office in reference to the shaft governor are of much interest, but time will not permit any extended reference to these

records. A few of the earlier patents are, however, considered of so much importance that drawings are submitted and quite full references are given. These early patents do not, probably, represent any practical application, but they are interesting as showing a complete understanding, not only of the theory of the shaft governor, but of methods of application to practical work.

The earliest reference which I have been able to

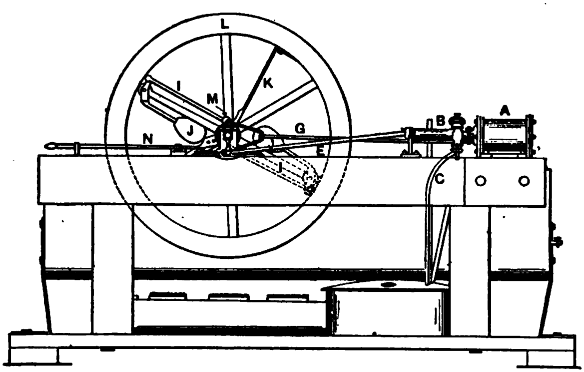

FIG. 5

find to the shaft governor is shown in a patent granted J. D. Custer, June 21, 1839 (Figs. 5 and 6). From these it will be seen that it consisted of two balls or weights symmetrically disposed in the fly-wheel and in gravity balance and pivoted to radial arms and connected by links with the eccentric in such a manner that the action of the centrifugal force would cause the balls to fly out, and this action would twist the

eccentric on its center so as to reduce the travel of the valve. The action of the centrifugal force was opposed by a flat spring. The drawing indicates a form of a governor which should have been of practical utility, but I have not been able to find, however, that the governor patented by Custer was ever put into

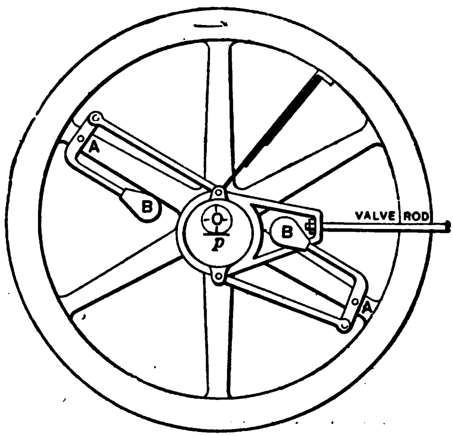

FIG. 6

practical use. It is quite certain that this invention did not produce any great change in the art of building steam engines, as the shaft governor seems to have been practically unknown for nearly a third of a century after this date.

The next governor patent to be granted was to Lewis Eikenberry, of Philadelphia, April 1, 1862 (Fig. 7). The patent was given principally for an improvement in variable cut-off valves, in which the valve motion

was regulated by use of a cam. The shaft governor shown was of peculiar type, in which the pivots or the arms to which the balls were fastened were in the plane of the revolving wheel so that the centrifugal force carried the balls into a position at an angle to the plane of the wheel and nearly parallel to the shaft.

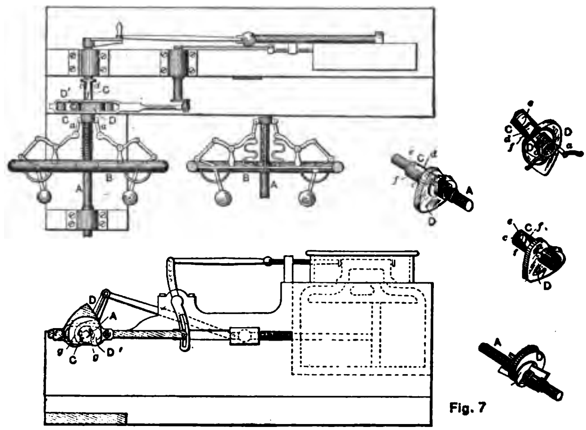

FIG. 7

This form of governor should have been efficient and effective, but it doubtless would have proved not practicable to apply in numerous cases. The next governor patent was granted to Joab H. Wooster, August 20, 1867, of which no picture is shown, of the same general type as that granted to J. D. Custer; in this patent, however, the eccentric was arranged so as to fit loosely upon the shaft and was connected to the governor in such a manner that it would swing past the center of

the shaft, thus changing the lead of the valve. The construction shown in the patent granted for this governor would probably have resulted in a partial success, but I have not been able to find evidence which would show whether or not this governor was put into practical operation.

The next patents in order, to which we will refer only by name, were as follows: Samuel Stanton, Newburg, N. Y., July 14, 1868; D. A. Woodbury, Rochester, N. Y., May 31, 1870, and also September 27, 1870. In the latter patent, which shows a governor used later in the well-known Woodbury engine, a distinct statement is made in the specifications regarding the effect of inertia on the parts of the governor, and the arrangement is made so that inertia, as well as centrifugal force, is employed for governing purposes.

The next patent in order was granted to Joseph W. Thompson, Salem, Ohio, July 15, 1872, and which, with a later one granted April 27, 1875, and still another on January 18, 1878, forms the basis of construction which has been used so long and with such excellent results in the Buckeye engine.

In chronological order patents were granted to John C. Hoadley, October 28, 1873, and March 17, 1874, for shaft governors, both of which were practically used on the Hoadley engine.

From this time on patents on shaft governors are exceedingly numerous and cover different forms of mechanical devices and different methods of application of mechanical principles. The improvements of a later date are generally of a nature which resulted

in simplifying the construction, reducing the number of working parts, lessening the friction and thus making the governor more perfect in its action.

The shaft governors can be divided into two classes with respect to the motion of the valve, namely:

Class I, in which the eccentric is rotated or twisted around the shaft. The travel of valve is changed without change of lead.

Class II, in which the eccentric is mounted on a disk with a center different from that of the fly-wheel and is swung in the arc of a circle across the center of the shaft. The travel of the valve is changed with change of lead.

For both the above classes of valve-gear the governor can be essentially of the same character, hence the above distinction does not necessarily indicate a structural difference in the governors.

Neglecting the difference of swinging or rotating eccentric, governors can be divided into three groups, depending on structural differences.

These groups are as follows:

I. Governors with two weights in gravity balance, as already shown in early examples in the Custer, Buckeye and Westinghouse governors.

II. Governors with a single weight in gravity balance, with eccentric and governor mechanism.

III. Governors with single arm in partial gravity balance which carries inertia weight, centrifugal weight and eccentric.

All the above classes can be operated so as to have regulation assisted or retarded by inertia and can

probably be connected to rotating or a swinging eccentric as desired.

A very good illustration of a shaft governor of the first class is shown in Fig. 8. The eccentric is mounted on a plate *G*, pivoted at *P* and is connected to *E B*, No. 1, and *E B*, No. 2, by connecting rods, in such a

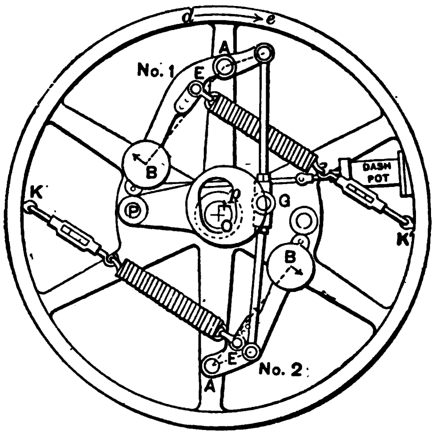

FIG. 8

manner that the action of centrifugal force in throwing the weights *B B* outward causes the center of the eccentric to swing toward the center of the shaft. The springs pivoted at *K* rock against the centrifugal force and hold the weights in a determinate position for each speed. The dashpot simply restrains the motion when too rapid and tends to prevent racing. There are numerous governors in this class.

Fig. 9 represents a notable illustration of a shaft

governor in Class II. This governor, although consisting of a single weight, is still in gravity balance. Its advantages over those in Class I are a less number of working parts, simpler construction and less friction.

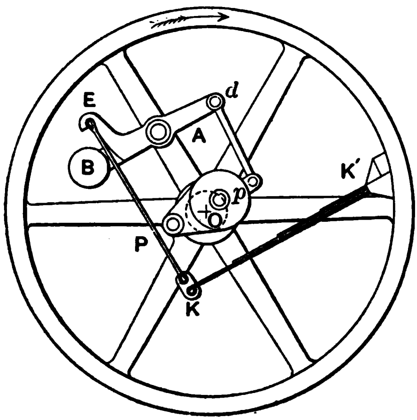

FIG. 9

The governor is used on the Straight Line engines and one or two others, and is the latest design of Prof. John E. Sweet.

Fig. 10 represents a governor in Class III. This governor was designed by different engineers and the patents are now owned by Mr. Frank Rites. It is now in very extensive use in the United States. This governor has a single moving part mounted on a single pivot. It is designed to take full advantage of inertia, and is so nearly in gravity balance that no bad results in regulation were ever shown by defects in balancing.

The friction in this governor can be reduced to a minimum and the results are great sensitiveness and wonderful regulation under adverse conditions.

The accompanying table gives a list of United States

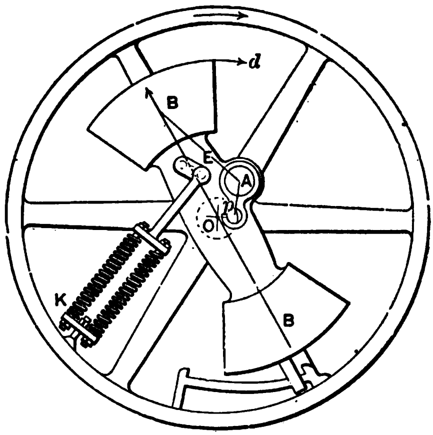

FIG. 10

patents for improvements in the shaft governor granted previous to 1880, in all only twenty-nine, of which five were granted before 1870, and twenty-five between 1870 and 1880. Since that date the patents have been numerous.

EARLY LIST OF U. S. PATENTS FOR SHAFT GOVERNORS

Patents Granted Prior to 1880

1839	June 21	1,179	J. D. Custer,
1862	Apr. 1	34,821	
1862	Apr. 16	38,055	L. Eikenbury, Philadelphia, Pa.
1867	Aug. 20	67,936	Joab Wooster, Strykersville, N.Y.
1868	July 14	80,025	S. Stanton, Newburg, N.Y.
1870	May 31	103,698	D. A. Woodbury, Rochester, N.Y.
"	Sept. 27	107,746	" " "
1872	July 16	128,986	J. W. Thompson, Salem, O.
1875	April 27	162,715	" " "
1873	Oct. 28	144,098	J. C. Hoadley, Lawrence, Mass.
1874	Mch. 17	148,560	" " "
1875	June 29	164,917	" " "
"	" 29	164,942	G. C. Suiss, " "
"	July 20	165,744	H. S. Maxim, Brooklyn, N. Y.
"	Aug. 31	167,225	Corbitt & Campbell, Milwaukee, Wis.
"	Sept. 21	167,835	J. Felber, St. Louis, Mo.
1876	Jan. 11	172,116	Hall & Whitteman, Hasma, N. Y.
"	Sept. 5	181,927	G. F. Ernst, St. Louis, Mo.
"	May 11	184,443	G. E. Tower, Annapolis, Md.
1877	Jan. 9	187,116	Cosgrove, Faribault, Minn.
1878	June 18	204,924	Thompson & Hunt, Salem, O.
"	July 30	206,500	H. Tabor, Corning, N. Y.
"	Aug. 6	206,792	C. B. Smith, Newark, N. J.
"	Sept. 3	207,607	D. O. Ladd, Chicago.
"	" 3	207,608	" "
1879	Jan. 14	211,309	L. H. Watson. "
"	" 14	211,335	C. S. Locke, "
"	Mch. 18	213,395	G. H. Cobb, Palmer, Mass.
"	Nov. 4	221,296	F. Fosdick, Fitchburg, Mass.
1880	May 25	227,967	C. V. B., San Francisco.
"	Aug. 17	231,228	W. Johnson, Lambertville, N. J.

The limits of this paper do not permit an opportunity for further discussion of the various forms of shaft

governor, or of its theory and method of action. The account of the development is imperfect, for the reason that the sources of information available were neither numerous nor exhaustive, but it is to be hoped that various members of the association will supplement the facts gathered together and presented in this short paper, with data relating to the development of the governor, while it is still fresh in mind.

There are many reasons for obtaining this information fully and in detail while there is an opportunity. Such investigation as made indicates that the shaft governor as we know it to-day is essentially an American invention, conceived, developed and perfected in this country.

The importance of this system of regulation is so fully recognized as to need no argument in its favor, and while at the present time the shaft governor is used only in an experimental way on certain classes of engines, yet the few experiments which have been performed indicate that its field is not limited to any great extent by speed requirements, and it seems reasonable to suppose that a period of development may extend its use to include not only all classes of steam engines, but gas engines as well.

The demand for close speed regulation came with the invention of the incandescent.

II

GENERAL DEFINITIONS AND RULES

BEFORE going further into the subject of governors it may be well to fix in our minds some of the definitions of terms used in reference to them, and already referred to in Chapter I.

Centrifugal force is that force which tends to fly from a center. A familiar illustration of it may be noted in swinging a weight, attached to a cord, about the head. The longer the cord the greater the force required to keep it revolving.

Centripetal force is force which always tends toward a center; the opposite of centrifugal force.

Inertia is that property of matter which tends to keep it at rest when resting, and, when in motion, tends to keep it moving in a straight line. It is this force which makes it difficult to start a heavily loaded wheelbarrow, and also to bring it to rest again when well under way.

Isochronal means relating to equal periods of time. This term is sometimes used in reference to shaft governors. The principal difference between the two general classes of governors, pendulum, and shaft is in the action of the forces which control them. In the pendulum governors there are the two forces, centrif-

ugal and gravity, which are equal at only one point of the operation of the same. In the shaft governor the force of inertia, or centrifugal force, is at all times opposed by an equal amount of spring-force. The weight-force increases as the weights move from the center, the spring-force also increases as the springs are extended by the weights.

When a governor is "sluggish," the speed falls far below its rating, and is not acquired again quickly, perhaps not at all. The weight-force is greater than the spring-force; the former must be decreased to get sensitiveness, and the latter altered to get the speed.

When an engine simply "speeds up" and must be checked on the throttle, either excessive friction in some of the parts exists or the spring-force is too great. Decrease the spring-tension to remedy this.

When an engine "races" or hunts," the two forces are unbalanced and are alternating rapidly in overcoming each other, causing the engine to alternate in speed within a certain range. Giving less tension on springs to decrease sensitiveness and changing weight to get the speed, is the remedy.

Racing may also be caused by friction of parts or other local troubles, as will be shown later in this chapter. There is, however, a noticeable difference between racing caused by over-sensitiveness and friction. When it is caused by the spring-tension alone the changes in speed will be rapid and even, within a certain range. When caused by friction the weights will stick on their inner position until the speed developed is so high as to throw them out with a noise;

or, when the engine is above speed, they will stick where they are until the speed is reduced enough for the springs to draw them back again.

The speed at which they will regulate, and the sensibility of the shaft governors depend principally on the following conditions: (1) Tension of springs; (2) the distance from the pivot where they are attached to the weight, or weight-arms; (3) the amount of weight; (4) the distance of weight from fulcrum.

Examples of, and Search for, Trouble

All of the well-known makes of shaft governors at the present date, of whatever class they may be, are thoroughly tested, regulated, and set by the makers, so that in the start they are turned over to the operating engineer regulating to within a certain range of percentage of speed called for, and are as perfect as they can be made. The difficulties that arise after being in service some time have a cause and a remedy.

Once a governor is perfected and running there is no reason why it cannot be brought back to that condition after it has been lost. If this fact is kept in mind, by perseverance the trouble will be readily found; often it is a very slight one, so small as to be easily overlooked. An engineer has been known to take a spanner-wrench and give the valve-rod gland a half turn to tighten it up, and so caused his engine to run away. Another had his engine, with a Sweet Governor, race because a single very small grain of gravel got between the band which connects the spring

and weight-arm and the weight-arm itself. Again a pinching cap on one of the fulcrum-pins or a slight burr on a valve-rod has caused trouble in a governor. The slightest thing should not be overlooked. Dry pins are often the seat of trouble; and a governor, to be properly attended, should be oiled as regularly as any other part of the engine, and once in a while all pins and bearings should be taken apart and cleaned.

When a search for trouble begins nothing should be neglected, from the governor-eccentric to the farthest edge of the valve in the valve chest. Disconnect the eccentric rod or rods, as the case may be, from the governor-eccentric, and remove or release the spring or springs from the weight-arm or arms.

Then move the weight-arms in and out on their travel from inner to outer positions. Most of the shaft governors made on engines from 5 H. P. to 1,000 H. P. are so counterbalanced that when thus operated one man should be able, on the smaller makes, to easily move the parts in and out with one hand, and, on the larger engines, with both hands, but he should never use a bar of any kind.

If they do not move so freely as to permit this the trouble is caused by dry or cut pins, pinching caps, bent rods or links making pins bind, pinching or dry eccentric-straps, or eccentric binding (in some instances between a bearing and governor-wheel hub) or sometimes gummed oil and grit cause it.

If the governor is free and in perfect condition disconnect the valves from the rockers or valve-rod slides,

as the case may be. Then look for dry surface of pins or bearings or slides, bent rods and other like conditions. This done, see that the valve stems are straight and true, and in line with their connections, also that their bearings do not bind and are not dry. See whether they are burred or worn small in stuffing box so that the packing binds it when pulled up tight, and whether the packing is old and dry.

Then look into the steam chest. See if the valve is set properly and if it leaks, or if the pressure-plate binds. Often an engineer forgets that proper valve setting is as essential as it is to have the governor free and well lubricated. An illustration of the fact that the valve setting must be carefully reckoned on is shown by the following experience:

A 500 H. P. cross-compound engine running condensing in a certain power house near New York City, began at one time to race and speed up very badly, and used much steam for no apparent cause. The steam pressure was 120 lbs. and the receiver pressure was from 45 to 70 lbs., which in itself showed something wrong with the valves, though the trouble was attributed to the governor.

This engine was vertical and had four gridiron valves to each cylinder, which allowed each valve to be set independently. The valves had small lap and the steam was admitted over the edges of the valves nearest the end of cylinder. An examination showed that the top steam-valve had been shoved up so that a late opening of valve occurred, and when the valve was supposedly lapped there was reopening of the

same on the opposite edges. This allowed the steam to blow through and on into the receiver, raising the receiver pressure and exerting a back pressure on the up stroke almost equal to the initial pressure on the opposite side of the piston. This made the H. P. cylinder inoperative, and the L. P. cylinder was doing more than its rating, thus unbalancing the engine and putting it beyond the control of the governor.

One turn on the valve-stem, drawing the valve into place, corrected all the trouble.

In one instance a large engine of well-known make ran for some time giving bad service — regulating badly. Finally it was discovered that the pressure-plates were so weak that they sprung in and pinched the valves while running, but were always apparently free when tested at other times. New and stiffer pressure-plates remedied this.

In cases where the direction of rotation of an engine is changed from running over to running under, or *vice versa*, the eccentric, and all governor parts, must be changed in their positions. The various makers give instructions for these changes, but the essential points to know in connection with quick changes are these: The pivoted ends of the levers should always lead, and the weights follow, the desired direction of rotation, and be so placed that when the weights move out the eccentric will be either advanced in the direction it will run for governors of the first class, Chapter I, or thrown across the shaft center in governors of the second class. Lack of a knowledge of this is sometimes a very serious source of trouble,

and these facts should be carefully stored in the mind, when a search for trouble begins.

At times it seems impossible to get enough spring-force to obtain proper adjustment of the governor, either from too long a spring or a weak one, more commonly the former. The remedy is to cut off one or two coils of the spiral spring until the desired effect is obtained. The best way to make such a cut is to spread the coils by driving a chisel between them and keeping it there until a score can be filed all the way, or at least three-fourths of the way around the springs; then remove the chisel from between the coils and finish the break with the chisel, laying the coil on an anvil or some heavy ridged surface. The flying coils, when they have parted from the rest, should be guarded against.

When we have a governor such as is described in the third group, Chapter I, we have the force of inertia to deal with in addition to the spring and centrifugal force.

In this type of governor, the weight on both the spring and free ends of the bar is inertia in effect, but changes of weight on one end has the opposite effect to the same change on the other end.

Changing the spring in this governor gives the same results as with all governors.

Changing the weight on the free end of these governor arms gives the same results as with the others.

A change of weight on the spring end of these arms gives the opposite effect to a like change on the other end. No radical change in weight of this class of

governor should be attempted without consulting the builder.

Sometimes, with the governor properly adjusted and free from friction, the engine will still speed up. This is caused by leaky valves or from insufficient steam-lap to cover the parts at all points of the engine-stroke, when the governor-weights are at the outer extreme of their travel. To test for this latter defect, remove the governor-spring or springs and block the weight-arms to their outer position, and then, while turning the engine one complete revolution, observe whether the steam edges or steam-valve covers the ports at all points of the revolution. If they do not, the valve setting must be changed to accomplish this.

The rules of action laid down in this chapter apply generally to all makes of shaft governors. Where radical changes are to be made, the builders should always be consulted, and the knowledge that each understands best how to operate his own special design of governor has impelled us to insert in the following chapters the rules of procedure, or instructions, of the builders, for use with each design named. In the study of the succeeding pages, the reader will note where these general instructions apply to the individual cases.

The two classes of governors as specified in Chapter I will be covered in these individual cases, but in the event of the operator not having an engine named individually in these chapters, the general rules of this chapter will no doubt cover the case.

III

ADJUSTING THE RITES INERTIA GOVERNOR*

THE inertia governor, invented by F. M. Rites, is now regularly used on engines made by considerably more than one hundred different manufacturers. It is thus the most commonly used governor for high-speed engines, and is already being adopted for use on slow-speed engines as well, either in place of the ball governor for Corliss valve-gears, or as a shaft governor for the large four-valve medium-speed engines now coming into general use. The principles governing its action, and the various ways of adjusting this governor to produce desired results, are of interest to every stationary engineer.

The Rites governor consists of a single piece of cast iron in the general form of a bar, mounted at right angles to the engine-shaft and carried on a pivot-pin parallel to the shaft. At a suitable point on this bar is provided a wrist-pin to which the valve-rod is connected, or if the governor is placed elsewhere than at the end of the engine-shaft, an eccentric is used instead of the pin. A spring opposes the inertia force of the bar.

* Contributed to *Power* by R. E. Cahill and S. H. Bunnell. This governor is in the second class of the third group, Chapter I.

The accompanying sketch (Fig. 11) shows the governor-wheel in outline, and the elementary form of the governor-arm. Observation will make it evident that the governor-arm, considered as two heavy masses *A*, *B*, will tend to overtake the fly-wheel if the engine-speed is reduced, as by increase of load, and to fall

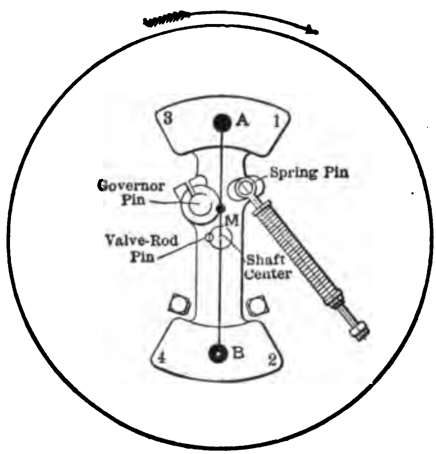

FIG. 11

behind the fly-wheel if the engine-speed is increased, as by decrease of load. It is also evident that the governor-arm, considered as a mass *M* located at the center of gravity of the whole arm, tends to swing in when the engine-speed is reduced, and out when the speed is increased. The arm therefore takes a position in which its centrifugal force balances the spring-tension (or as nearly that position as the arm stops will allow),

and moves relatively to the engine-shaft forward and inward if the engine-speed is decreased, and backward and outward if the engine-speed is accelerated.

The valve-rod pin (or the crank of the eccentric if that is used) is located nearly on the line from the arm-pivot center to the shaft-center, and distant from the shaft-center by the lap of the steam-valve when the governor-arm is in full-speed position. In practice, the governor is keyed to the shaft so that the arm-pivot pin is a little ahead of the center line of the engine-crank when at full speed. To prevent running over speed when without load the steam-lap must be great enough to give practically no opening when the governor-arm is in full-speed position, which means zero lead in this position. As the arm swings in, the lead increases, but not enough to give a proper lead in the usual running position unless the governor is set a little ahead, as described. The corresponding disadvantage is an excessive lead at late cut-offs.

The governor is designed by the engine builder in accordance with certain emperical rules developed by Mr. Rites from extended experience. It should have power enough to actuate the valves of the particular size of engine for which it was designed, and should only need adjustment in some of the several ways provided in order to meet the special requirements of any particular case. The first step in correcting faulty regulation of an engine is to determine the speed under a small load, say one-fourth of the rated load of the engine. If the speed is steady under small changes of this load, but too slow, tighten the gov-

ernor-spring; slacken the spring to decrease the speed. If the spring is not strong enough, so that screwing up further has not the effect of raising the speed, or if the spring is stretched to the limit of space allowed, one or more coils may be cut off, or any attached weights removed from the short end *A* of the arm. If the speed is not steady, but changes irregularly without corresponding change in load, look for trouble in the pivot-pin bearing — lack of oil or a cut and scored pin or bushing, and correct this first.

Next increase the load and observe the speed of the engine. If it drops more than desired, try setting the spring-pin farther toward the governor-arm pivot along the slot provided, or remove any attached weights from the end *A* and reduce the spring-tension; or add a small weight to the end *B* of the arm on the spring side, or both. Moving a weight on the end *A* from 1 to 3 has a similar effect, but in less degree.

It sometimes happens that the drop in speed cannot be overcome by the usual methods of weighting. In such cases, first making sure that the lap of the valve is sufficient, look for a hard-running valve, which, at full stroke, pulls excessively on the governor, springs the rocker-arms and connections, and by the combinations of fault causes the speed to drop. If possible, keep the load steady while counting or otherwise observing the speed. If the speed does not drop somewhat from light load to full load, the governing will probably be unsteady under quick changes, and the spring-pin should be moved out in the slot, or weight added to the short end of the arm on the spring

side. After any such change the speed will have to be brought back to the desired rate by adjusting the spring, as at first.

Next try the speed with all load off the engine, if that condition is ever likely to exist in the plant. If the speed rises considerably, the steam-valve leaks, or the steam-lap is insufficient to cover the ports entirely when the governor-arm is in the full-speed position. It is often found that a valve which apparently has the proper amount of lap will open slightly as the piston advances and allow the engine to run considerably over speed when the load is thrown off. Condensing engines will almost invariably run considerably faster without load, and it is best not to attempt to keep the no-load speed down to the exact figure, as the increased lap necessary makes the lead in the running position deficient. If the valve is decided to be too short, it is often easiest to make an offset-pin for the valve-rod, and put this in place of the regular pin in the governor-arm so as to decrease the throw at minimum travels, and thus save buying a new valve. Careful observation of the speed of the engine under different loads, and successive adjustments in the manner described, will soon bring the engine to the desired condition.

In adding weights it is well to bear in mind that a change in the weight of the governor-arm as a whole is not what is wanted, but a change in the distribution of the weight. If you find yourself about to add a weight which will act exactly opposite to one already in place, try taking off the other weight first; perhaps

none is required. If the desired regulation has been obtained by a combination of weights on one or both ends of the arm, experiment will usually prove that the same result can be secured by a single weight properly placed. It is merely a question of balancing the centrifugal force of the governor-arm against the tension of the spring. If these are exactly balanced at all points there will be no permanent change of speed from no load to full load, which is sometimes a desirable condition and is easily attained by the inertia governor; or the weight and spring-pin may be arranged so that the balance will vary at different points of the movement, the arm requiring a greater speed to hold it out against the extreme tension of the spring than to balance the spring-tension in other positions, giving an increase of speed as load decreases. By overbalancing the governor, an engine could be made to run much faster with load than without, but for safety and reliable running the full-load speed should be nearly two per cent. lower than the no-load speed.

The adjustment of speed to load as described depends on the centrifugal effect. Steadiness under change of load depends on the inertia effect, and is next to be considered. When the load is suddenly increased, the consequent checking of the engine-speed allows the governor-arm to run ahead of the wheel, carrying the center of gravity and lengthening the cut-off. If the fly-wheel is sufficiently heavy and the inertia effect of the governor-arm great enough, the engine-speed may drop only slightly. But with a free-moving governor the arm is likely to swing too

far, resulting in too late a cut-off and an increase of speed after the momentary drop as the load first came on, followed by a swing the other way as the engine overruns the governor-arm, and so on. These swings are quite regular, and very clearly shown by the voltmeter on a direct-current unit. If a sudden change in load produces two or three long swings before the engine finally steadies itself, try adding a weight to the long end of the arm, on the line through the centers of the pivot and the shaft. One swing is to be expected, but the engine should be so regulated that it will swing once up and back to the correct figure, never passing the normal speed twice for one change of load. If the speed changes too much at first and comes back too slowly, extra weight on the long end *B* of the arm is probably needed, as in the other case.

The most troublesome condition is irregularity. Engines are sometimes found to vary speed unaccountably, perhaps suddenly, and at odd intervals. Sticking at the pin is a common cause of this, but too free a pin may possibly allow the governor to float under insignificant impulses and produce a similar effect. The governor-arm is unbalanced against gravity, and if the engine is run at too slow a speed it may fall forward somewhat during half the revolution and backward during the other half, making the cut-off too long on one end, or irregular in successive strokes. Sometimes the gravity effect combines with valve-rod friction or inertia and makes the motion kick the governor so that the valve-gear moves with peculiar jerks. A simple brake, as a piece of flat spring bear-

ing on the arm, or a dash-pot, may be the easiest means of controlling this. A large, stiff governor-pin introduces just the necessary element of friction to make the governor stable, and is thus desirable for other reasons than strength.

A common cause of complaint with large governors is hammering on the stops in starting or shutting down the engine. This can usually be overcome by moving attached weights and noting whether hammering is inrecased or diminished. Usually the proper change is in the direction of adding weight on the spring side of the arm and increasing the spring-tension, though it may be necessary to add weight at both ends. It is a peculiar fact that friction in the valve-gear operates to help the governor-spring, so that an engine may be speeded up several revolutions by excessively tight valve-stem packing or any similarly acting cause. It is well to look over the valve motion as a possible cause of any unaccountable change of speed. If a brake is used on the governor and is set up too tight, it may cause continual changes of speed through its action in checking the governor-arm as it swings out or in, and so preventing the arm from floating gradually to the proper position.

It may be necessary to adjust the governor with no other data than what can be learned by watching the switchboard meters while the engine runs in service, and applying the proper remedy for the apparent fault on the occasion of the next shut-down. It may take an hour's careful watching to make sure regarding the real action of the governor; for the only sure way is

to wait for the load to change as desired and remain constant long enough to give the engine time to settle to a steady speed, and repeat the observation until the exact speeds under several different loads are ascertained.

In conclusion, before altering a Rites' governor the engineer should make sure that the main pin and its bushings are free and properly lubricated, and that the valve has sufficient lap and runs freely. If the arm is heavy enough to drive the valve, see whether the desired governing effect can be produced by adjusting the spring; also avoid adding unnecessary weights and the consequent overstraining of springs, bushings and pins.

IV

THE BUCKEYE ENGINE GOVERNOR AND ITS ADJUSTMENTS

THE governor of this engine of which (Fig. 12) is a cut, comes in class 1, group 1, as specified in Chapter I. The following instructions are for its adjustment.

NAMES OF PARTS

The following names are given to the several details of the governor for convenience of reference.

The levers or *weight arms a a* will be called *levers* hereafter for convenience.

The weights A A are clamped on the levers.

The lever pivots b b are studs, secured to arms of the containing wheel on which the levers move freely.

The links B B couple each lever to ears on the sleeve of

The Governor eccentric C, which is free to turn on the shaft and is turned about 90 deg. on the shaft by the outward movement or *expansion* of the levers to the outer extreme of their range of movement.

The main springs F F are of tempered steel wire. They are anchored adjustably to the rim of the containing wheel by means of

The tension screws c c by which the tension is adjusted.

The spring clips d d are clamped on the levers *a a* and are provided with slots or *eyes* into which the

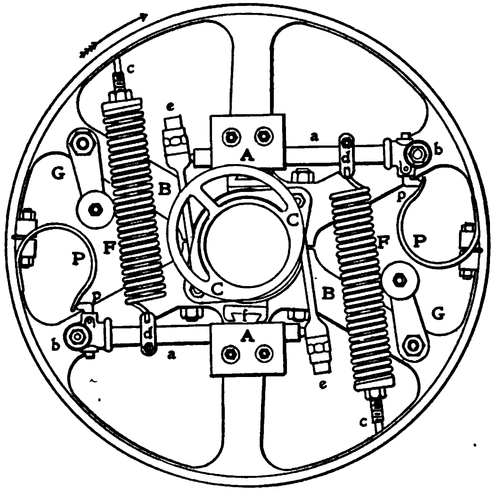

FIG. 12

springs *F F* are hooked. They may be moved along the levers and fixed in any position within narrow limits.

The lever stops f f are blocks of wood on which the levers rest when not expanded. They are held in dovetail recesses in brackets bolted to the containing wheel.

The outer lever stops e e are cylinders of wood fitted to sockets in the outer caps of the links *B B*. If the levers expand violently they strike the inner surface

of the containing wheel rim, but with proper adjustment they seldom or never touch the rim.

The auxiliary springs P P are introduced *to help the levers out* during the first half of their outward movement, when the main springs *F F have enough tension to give close regulation at light but varying loads.* Without them and with such tension the governor would race with standard or heavy loads.

The guide rollers G G are introduced in most high-speed engines to restrain the springs from bowing outward from centrifugal force. They are most needed when speed is 250 and upwards, and when the spring clips *d d* are short. [In one or two sizes clips of different lengths have been used.] The trouble that called for their use was due to the change in direction of pull on the clips in consequence of such bowing, and which caused racing when the amount of tension called for by calculation was applied.

Table of Governor Data

The governors are made in six sizes, numbered 1 to 6. The "diameter of wheel" will serve to identify any one the data of which may be wanted.

	Number of Governor	1	2	3	4	5	6
A	Diameter of Wheel (inches) . .	24	32	40	48	54	66
B	Spring leverage " . .	$4\frac{8}{18}$	$5\frac{1}{16}$	7	$8\frac{1}{2}$	$9\frac{1}{2}$	12
C	Weight leverage " . .	$8\frac{3}{4}$	11	14	17	19	24
D	Initial spring tension " . .	$2\frac{1}{2}$	3	$3\frac{7}{8}$	$4\frac{1}{4}$	$5\frac{1}{4}$	$6\frac{1}{4}$

DATA FOR WEIGHT CALCULATIONS

E	Effective wt. of levers (lbs. oz.)	$\frac{2}{12}$	$\frac{6}{12}$	$\frac{9}{12}$	18	20	32
F	Assumed wt. orbit (ft. diam.)	1	1.25	1.5	2	2	2.75
G	Resultant spring tension (in.)	3.25	4	5	6.25	6.5	8

Explanation of the Table

The diameter of wheel is given as before explained for identification. When making calculations or referring to data for any purpose, use only those under the given diameter which agrees with the wheel of the governor under consideration.

B. *The spring leverage is the distance* from the centers of the pivots of the levers to the centers of the eyes of the spring clips. It is adjustable, but the amount given is that on which all calculations are based. It is fixed at one-half of the weight leverage (*C*) for convenience of calculation. It is also very nearly all that can be had in each case, for reasons to be made clear presently, but it can be diminished in all cases.

C. *The weight leverage is the distance* from the centers of the pivots of the levers to the point where the whole effective weight of the levers and attached weights is assumed to be concentrated and which comes about central over the "lever stops."

D. *The initial spring tension, is, as nearly* as can be determined theoretically, the maximum tension that can be applied without racing, the "Spring leverage," (*B*) being as given and the auxiliary springs applied

and properly adjusted. It is more than could be carried in the absence of the auxiliaries, unless with very careful adjustment, and other conditions favorable. (See 71 and 129.)

E. *The effective weight of a lever is the* weight of an unweighted lever with spring clip in position to which is added one-half of the weight of a link (B, Fig. 12). The weight is found by resting the lever on the scales at the distance from the pivot given as the limit of

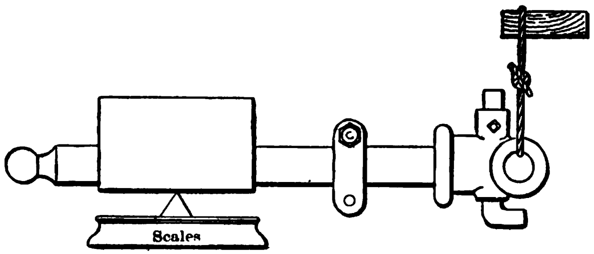

FIG. 13

the "weight" leverage (C) while the pivot is supported independently of the scales. (See Fig. 13.)

F. *The assumed diameter of the orbit of* the weights is an orbit somewhere within the range of movement of the levers, so chosen that its diameter will not contain inconvenient fractions of a foot, as it is assumed solely for purposes of calculation. The diameter assumed is immaterial provided the next item (G) is correctly deduced from it.

G. *The resultant spring tension is the* initial tension (D) augumented by the additional tension that would be imposed on the spring by moving the levers out-

ward till their centers of force reached the assumed orbit. (Neither this nor the initial tension can be given exactly for all cases, as the latter depends somewhat upon the position of the actual center of force, which varies in distance from the center of rotation, as the levers are heavily or lightly weighted, being farthest from the center with heaviest weights. But both weight and spring leverages are sufficiently adjustable to enable the desired speed to be attained when the calculated weight is attached.)

Use of the Table

To calculate the weight required for a given speed.

In addition to data furnished by the table, the force of the main springs in pounds per inch of tension will be needed. This will be generally found stamped on the cast heads of the springs; if not, the springs may be hung up and weighted till extended one, two or more inches, when the weight used divided by the inches extended will give the force, which for convenience may be represented by the symbol "f."

The first step in the calculation is to find the centrifugal force of each pound of weight revolving in the assumed orbit (F) at the given speed, which may be represented by "S." The desired force being represented by "cf" the formula will be, cf $= S^2 \times F \div 5870$.

Next we wish to find the spring force at the point of weight leverage (C) and in the assumed orbit (F) which we will represent by "sf." The weight leverage being twice the spring leverage the formula will be, $sf = f \times G \div 2$.

Then sf ÷ cf = the theoretical total weight,* from which the item E is deducted, leaving the amount to be added to each lever.

Example. Find weights for No. 3 governor, speed (S) 180, spring force (f) 76 lbs. per in. The assumed orbit (F) 1.5 ft. and the resultant tension (G) 5 in.

For the benefit of those not familiar with formula we will give the rule arithmetically.

The force per lb. (cf) is found as follows: *Multiply the square of the desired number of revolutions per minute by the diameter of the orbit in feet* (F) *and divide by the constant number* 5870.

Thus $180^2 \times 1.5 \div 5870 = 8.28$ lbs. very nearly, that is, each pound in the given orbit will exert 8.28 lbs. centrifugal force.

Then the spring power 76 multiplied by the resultant tension (G, 5 in.) will give the total spring force at the *spring leverage,* the half of which will be the spring force at the *weight leverage.*

Thus $76 \times 5 \div 2 = 190$ lbs. Then $190 \div 8.25 = 22.94$ lbs. total weight required. Deducting one-sixth from this as per note below it becomes 19.12 lbs. or 19 lbs. 2 oz. Then 19 lbs. 2 oz. — 7 lbs. 14 oz. (E) = 11 lbs. 4 oz. to be added to each lever at point C.

For other speeds, other things equal, only the first

*Owing, however, to several disturbing influences, namely: — the centrifugal force of the spring itself; the friction of cut-off valve which acts in a direction to aid the spring, the inertia of valve and valve gear, the friction of yoke on eccentric and of eccentric on shaft, as well as friction of pivots, — a correction must be applied to this theoretical total weight. Experience shows that five-sixths of this amount is usually enough.

part of the calculation, finding the cf, needs to be gone over again.

AUXILIARY SPRING ADJUSTMENTS

The function of these springs has been already explained.

They were first applied in the latter part of 1884, for the purpose of securing the exceptionally close regulation required for electric lighting.

As their adjustment cannot be perfected till after the engine is started, the shop adjustment (which is the best that can be made by a general rule) may in many cases require to be changed in order to secure the best results.

The test of perfect adjustment is, of course, *close regulation at all loads without racing at any load, and prompt response to changes of load without objectionable change of speed, momentary or permanent,* but by carefully observing the performance of the engine *at starting* the engineer can with a little experience tell almost as well when its adjustments are perfect and what changes may be needed, as by the test of regular running. But to do so he must first familiarize himself with the appearance of the governor sufficiently to be able to tell the moment the levers begin to expand as well as how quickly they do so, and to detect any irregularities in their outward movement.

Making white or bright colored spots on the weights with chalk, paint or paper will greatly assist such observations.

Perfect adjustment may be recognized by the following

performance: *On starting the engine gradually* the weights will not start outward till the proper speed is very nearly reached — so nearly so that the lack of it is not noticeable — when they will expand quickly but not violently, or so as to strike the outward stop; going out, however, nearly their full range, when if the load driven is heavy enough to require less expansion, they will promptly return to the requred position.

If, however, they make a few slight oscillations to and fro past their position no harm will result, if only they always settle in good time. *Very close regulation* requires that the *equilibrium shall be at the very verge of instability,* a proposition that will be recognized by all who have thoroughly studied the subject, *as true of all centrifugal governors.*

Auxiliaries too weak. The performance in such case will be the same in kind as though they were absent entirely, though more moderate in degree. On starting, the engine will run above its proper speed before the levers will expand, when they will fly out violently, and stable regulation will be possible only with loads so light as to regulate at one-fourth stroke cut-off or earlier, that is, such as require the levers to act only in the outer half of their range of movement. At heavier loads, the governor will race continually.

Auxiliaries too strong. On starting up the levers will start out at noticeably *less* than proper speed and expand gradually as speed increases till the limit of the follow of the auxiliaries is reached, when if they are much too strong, the expanding movement will stop a little till proper speed is reached, when they

will finish their expansion with proper promptness. The regulation will be the same as in both previous cases when the load is too light to bring the auxiliaries into action, but with heavier loads the speed will be slow in proportion to the undue strength of the springs. At maximum load, that is, just sufficient load to bring the levers to their inner stops, the speed will be reduced to about what was required to start them out.

In all of the three foregoing cases the tension of the main springs is assumed to be what it should be with the auxiliaries at their best adjustment.

To enable the engineer, whose engine is without them, to judge whether and to what extent his regulation would be improved by their application, we give a description of a performance capable of improvement, assuming the tension of the main springs to be all that can be carried without racing at any load, which *is always less than will be needed when auxiliaries are applied.*

Best regulation without auxiliaries. At starting the levers will not start out till proper speed is nearly reached (as per 81), but they will expand quickly *only in part;* from about mid-movement outwards the expansion will go on only as speed increases, requiring a greater increase of speed to expand them to near their outer limits than that which sufficed to expand them through the inner half of their movements.

The regulation in such case may be good at all loads requiring one-fourth stroke cut-off and later but with lighter loads, requiring earlier than one-fourth stroke

cut-off, the speed will vary much more with a given change of load than with heavy loads.

The strength of the main springs is however a factor of some influence in determining the degree to which the foregoing performance falls short of perfect regulation. The stronger they are the closer the regulation, throughout the whole range, that can be had without the help of auxiliaries.

From the above it might appear that, given main springs strong enough, the auxiliaries might be dispensed with entirely, which is true in some cases; yet the strength necessary to obtain that result in all cases would impose such severe pressure on the lever pivots that the resultant friction would interfere to some extent with fine regulation.

It is a matter of many year's experience that the closest and most sensitive regulation possible requires that the forces in equilibrium within the governor be not so great but that the work imposed on it will very slightly disturb the equilibrium at each stroke, so as to *overcome the static friction of the joints and eccentric sleeve,* and enable the parts to adjust themselves to the load requirements without having to await an objectionable change of speed to do it. And when the forces are weak enough to be thus sensitized there is left a small margin of improvement to be effected by the auxiliary springs.

Applying auxiliary springs to old engines. As before stated they were not used till 1884, and although many have been since applied to engines built before that time, there are still many running without them.

The indications for their use have been already given. but when such indications are present, the main springs should be examined, and if of the kind now made, namely, with hooks on one end only, the other being closed with a cast head threaded for the tension screw, and if figures can be found stamped on the heads we would recommend the parties to advise us as to the power of their springs, and if the requirements for regulation are not exceptionally exacting, it may happen that a stronger pair of springs, with required weights, will be better on the whole than the application of the auxiliaries. (Our records, however, give the spring force in all engines shipped since and including October 9, 1882.)

To apply auxiliary springs. This should be done in all cases where *best possible* regulation is desired, as is generally the case with electric lighting plants or engines to be used wholly or partly for that purpose.

We can send the springs, bolts and fingers adapted for use with existing levers, as shown in Fig. 14, or we can at not materially greater cost send new levers with fingers fitted as shown in Figs. 12 and 13.

To fit to existing levers, $\frac{7}{8}$-in. or $1\frac{1}{8}$-in. holes should be drilled through the cast heads of the levers as near the pivot holes as possible without danger of breaking into them, and at right angles to both the levers and the pivot holes. The surfaces around the holes at each side should be faced by chipping, or better, "*rousting*" if a machine-shop is in reach, — to receive the lock-nuts shown, and give them a fair bearing.

The springs are bolted to the rim as shown in Fig. 12, the angular position selected being such that the fingers will just catch with certainty at their *shortest reach.*

When the springs are secured in position, the eccentric should be turned forward till the fingers leave

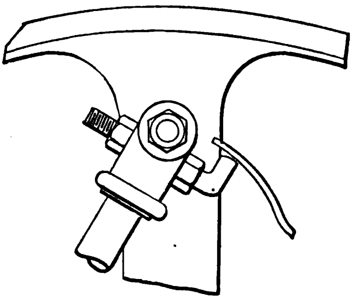

FIG. 14

contact with the spring, which should happen when the levers are about half way out or a little more. If they leave contact too early or too late, they should be taken off and bent outwards or inwards, as required, till they follow as above. They are not tempered and will not break.

Add tension to the main springs till regulation is as close as desired between lightest and medium or standard load.

Correct the speed by adding to weights, shifting them from pivots, or diminishing spring leverage, or by two or more of these adjustments.

Compare performance with foregoing descriptions. If

the springs appear too weak give the fingers more reach. If too strong (as will be more likely the case) take one of them off. If still too strong, grind one weaker and use it alone. Grind liberally and fearlessly, for if it is made too weak the other can be similarly ground and applied, or finger given more reach. *Strength is easier got than weakness*, yet the lesson is more instructive if the point of insufficient strength is reached and carefully corrected.

To Change Speed

For any considerable change of speed the weights should be changed, the proper weight for desired speed being found by rules already given.

Slight changes, however, can mostly be made by adjustments, of which the following are preferable:

To Increase Speed

A. *Increase of spring tension may be tried*, and if when the desired increase of speed is effected in that way the regulation remains sufficiently stable, *i.e.*, free from tendency to race at any time, the correct adjustment has been made, and the regulation will be closer than before. But if the tension has been made what it should be — all that can be carried without racing — it cannot be increased, in which case

B. *The weights may be shifted* towards the pivots of the levers, provided they are not already as far in that direction as permissible. [They should not be far from central over their stops in that direction.]

C. *The spring leverage may be increased* by slipping the spring clips farther from the pivots, provided the link heads are not thereby caused to strike the springs at mid-movement, as may be tested by turning the eccentric forward past its mid-position. A slight interference so detected will not matter, as when running, centrifugal force will bow the springs outward, if not too closely restrained by the restraining rollers now applied in many cases to high-speed engines.

When the spring leverage is increased, an increase of spring tension equal in amount to about one-half the increase of leverage becomes admissible as *the maximum possible tension is a certain portion of the leverage* (not to the same in all cases exactly, however), not a certain absolute amount.

To Decrease Speed

From the foregoing it will be evident that

A. *Spring tension may be reduced* if leverage is reduced twice as much at same time, without introducing greater speed variation, as *reducing spring tension alone would do.* But this adjustment should not be resorted to for any considerable change of speed, as it introduces objectionable weakness in the governor.

B. *The weights may be shifted* farther from the lever pivots, if not already so far from their normal position in that direction as to render any further shifting objectionable, though no trouble is to be apprehended so long as they are clear of the links in all positions.

C. *Spring leverage may be reduced* without con-

current *reduction of spring tension,* provided the latter is not at a maximum. The test of that is the *performance;* if *racing* is not induced, the adjustment is admissible.

To Reduce Speed Variation

A. *Increase spring tension,* if possible without danger of overstraining.

B. *Diminish spring leverage,* if not already as much as advisable less than normal.

C. As A increases speed and B reduces it, a certain combination of both together, that is, about twice as much reduction of leverage as increase of tension will accomplish the desired result without change of mean speed.

If racing results, note whether it occurs at heavy loads only, or at all loads. If at heavy loads only, *make the auxiliary springs follow noticeably more than half of the lever movement,* and try first with one of them removed, and if one alone appears too weak, try greater reach of finger.

If all adjustments of the auxiliaries, however, fail to cure the racing, it may be concluded that the previous adjustments A B have been overdone.

Racing from All Causes

Enough has been said to make the engineer perfectly familiar with the fact that racing may *always* be stopped by reducing spring tension, increasing spring leverage, or both, and nearly always by increasing the

force, or prolonging the follow of the auxiliary springs. But cases may arise when none of these adjustments should be made. Such is presumably the case when it appears spontaneously under adjustments that have previously given satisfactory regulation, and also when the tension is not in excess of that given in the table, and it refuses to yield to any moderate auxiliary spring force or follow, and particularly if, when cured by auxiliary spring adjustments, the speed variation with load changes is objectionably great.

In such cases *undue friction* will undoubtedly be found to be the cause of the trouble. It may be in the lever pivots, the ball and socket joints of the links or the loose eccentric on the shaft, one or more of these bearings; and may be caused by over tightness, lack of oil, rust or gum. Only the ball joints can be tested without taking the governor apart, the play at the necks of the balls, allowing the links to be slightly rotated back and forth, and when this can be done easily they are free enough. The lever pivots can be tested by taking off the retaining nuts and washers of the studs and slipping the levers partly off, when the condition of the exposed surface will be apparent, and the needed remedy (cleaning and oil) readily applied. But to test the condition of the eccentric bearing perfectly, the eccentric should be both unstrapped and disconnected from the levers so as to be rotated freely on the shaft. If dry or gummed, it may be simply oiled with or without preliminary doses of turpentine or kerosene, but if this fails to eliminate all sticking points, the governor should be slipped back or taken

off to allow the eccentric to be moved aside (the larger sizes are made in halves and hence can be removed) when any brusies or tight points can be discovered and corrected.

New engines will seldom require such treatment unless the eccentric has been too closely fitted, but older ones, especially after standing some time, or the use of gummy oil, may need it.

The kind of racing caused by friction is, however, noticeably different from that due to over tension or insufficient auxiliary spring force, as follows: when caused by friction the levers will expand and *stick* in that position till speed falls more or less according to the amount of friction, when they will drop in and again stick till the speed increases sufficiently to again expand them, and so on. Apparent sticking on the *inner* position is not to be taken as evidence of friction, since that will happen with insufficient auxiliary spring force; but nothing but friction will cause *dwell* in the outer position, during considerable change of speed.

Over-packing the cut-off stem will disturb the equilibrium of the governor and cause irregular action, but not usually racing as above described, but rather irregular flopping in and out of the levers.

The cut-off stem should not be packed with any of the hard kinds of packing, and such soft kind as may be used (candle wick is as good as anything) should be renewed often enough to avoid the necessity of screwing it up so tight as to cause friction enough to disturb the governor and wear the rod out.

Undue friction of the eccentric straps, whether from

lack of oil or too light adjustment will sometimes cause racing, accompanied by *acceleration of speed*, much as though the spring tension had been considerably increased. Some acceleration of speed always results from this cause, even when racing does not, and the same is true, though to a less extent of undue friction of the cut-off valve, its stem packing or its rocker-shaft and pins; and, as no other accidental change (except the slipping backwards of the governor wheel) can cause acceleration, when that symptom appears attention should be at once directed to the conditions of the parts named.

The difference between the effects of undue friction of the above-named parts, and of the working parts of the governor and the eccentric on the shaft should be well understood by the engineer. Friction of the latter parts may be called *static* friction, as it tends to hold the parts concerned *stationary*, relatively to the shaft and wheel, as against the movements required for cut-off variation, *in both directions alike*, while friction of the other parts named tends to pull the levers of the governor *inwards*, hence it may be called *dynamic* friction, or since inward pull on the levers is a *centripetal* action, like that of the main springs it may be more descriptively called *centripetal* friction.

From the above it will be understood that it is the *static* friction that most tends to cause racing when in excess. Of the parts concerned in producing *centripetal* friction only undue friction of the eccentric straps will cause racing, because that of the other parts,

being absent at the dead centers of the eccentric movement, is too intermittent to cause any other disturbance than that already described in Sec. 116. The eccentric strap friction, on the other hand, is tolerably constant, and consequently acts like increased spring tension.

It will be seen from the foregoing that there are *three* frictional effects going on in the governor, namely, the *static*, the *constant centripetal*, that of the eccentric straps only, and the *intermittent centripetal*, that of the cut-off valve, its stem and the joints and bearings of its gear.

The *static* and the *intermittent centripetal* frictions, when normal, counteract each other's bad effects, so that regulation can be, and mostly is, *as sensitive as though all parts were entirely frictionless*. Thus, the former prevents the latter from jerking the levers inward to an objectionable degree each stroke, yet not so effectually but that it (the static friction) is overcome and the eccentric turned by a minute amount at each jerk, while it recovers its position by contrary movements between jerks. The static friction being thus overcome four times in each revolution, in each direction alternately, is *practically neutralized* leaving the governor entirely free to respond instantly to all changes of load or pressure.

The *Constant Centripetal* effect of the friction of the eccentric straps has no material effect on the *sensitiveness* of the governor; it only slightly increases speed, other things equal.

But this centripetal effect is not uniform through-

out the range of movement. It is the greatest at the extremes of the range where the angle formed by the links B B (Fig. 12) with a line joining the pins in the eccentric ears is acute or obtuse, and least near the middle of the range where it is a right angle. From this fact results the need for the auxiliary springs. The entire theory of the matter need not be explained here; — the leading lacts being sufficient for those who do not care to study the subject exhaustively.

The auxiliaries permit the spring tension to be adjusted to the requirements of the outer half of the range of movement, while they prevent the tension from being in excess during the inner half, as it would otherwise be.

The Theory of Spring Tension

The force of a spring increases in direct proportion as it is bent (by extension in present case, or in whatever way it is acted on), and the centrifugal force of a body in like manner increases in direct proportion as it moves farther from the center of motion, the number of revolutions per minute remaining constant.

Consequently, in the absence of all disturbing causes, if in a governor of the kind in question, the spring tension be made such that if the lever be moved inwards till its center of force reaches the center of motion, or a line joining its pivot and the center of motion, in other words, its *zero of centrifugal force,* it (the spring tension) would reach its zero at the same time, the two forces would increase in the same ratio (at a constant rotative speed) as the lever moved

outward, and consequently the speed would be the same at all points in the range of movement; in other words, the regulation would be *isochronous*.

But suppose the tension to be less than this, so that as the lever moved inwards the zero of spring force would be reached before that of centrifugal force, then, as it moved outward the spring force would increase more rapidly than the centrifugal force at a constant rotative speed, so that a constantly increasing speed would be required to keep the forces in equilibrium, and the number of revolutions the speed would have to increase in order to carry the lever outwards through its range of movement would be the extreme measure of the governor's variation. Thus, if 100 revolutions in a given time be required to start the levers outward, and 105 in same time to expand them to the outer limits of their range, the extreme variation would be 5 per cent., which would be tolerably close regulation, seeing that in practice the changes of load and pressure seldom cover more than half the range.

To Obtain Closest Possible Regulation

Although enough has been said in Secs. 80 to 100 to cover the entire ground, yet a concise rule in this place will be convenient.

1st. Give the main springs all the tension that can be carried without racing at any load from nothing up to near quarter cut-off, as nearly as can be judged. If indicator cards can be taken to show range of cut-off the test will be far more intelligible. If tension

cannot be given as desired, on account of fear of overstraining the springs or lack of room at the tension screws, the spring leverage may be reduced a little; but in some way get tension or its equivalent, till the regulation within the above range is as close as desired.

2d. Count the speed at as heavy a load as can be applied with certainty that the weights do not touch their stops. If indicator cards can be consulted, apply load till it shows about half-stroke cut-off. Generally, as the auxiliaries are adjusted at the works this speed will be too slow. If it is more than 3 or 4 per cent. slower than the light load speed, reduce the auxiliary spring force till the speed is brought up as near the light load speed as desirable. Reduce first by diminishing the finger* reach as much as possible, and if this fails to bring the speed up as desired, take off one of the springs. If still too slow, grind the remaining one weaker unless it is found that it follows three-fourths of the distance out or more, when it may be sprung together a little, but in no case so much as to reduce the follow to one-half the movement. It should be noticeably more than half, unless less is finally found to be better by actual comparative test. If now no racing occurs at any load, the adjustment will probably be as perfect as desired, though a count of as many intermediate loads within the range of the action of the auxiliaries as possible may reveal some irregularities worth while correcting. For instance, if on counting under a series of loads from

* The "fingers" are shown in Fig. 12, at *p. p.* Reference to them in proper place was inadvertently omitted.

heaviest down, the gain of speed as load diminishes is found to be proportionately too rapid at first, the auxiliaries should be made to follow out farther, but at the same time weakened sufficiently to prevent making the half-cut speed any slower, as would happen if the spring or springs were simply opened to prolong the follow.

But the last correction is unnecessary if no signs of racing appear; the regulation within the proper working range will be closer without it, but bear in mind that with too short follow the light load regulation may be perfect and the half-cut speed not objectionably slow, yet at certain loads between half and quarter-cut it will race, or come too near it for perfectly satisfactory performance.

To Change the Direction of Motion

The main eccentric follows the crank about 60 deg.

The governor, however, must be taken apart entirely; the lever pivot studs *b b* removed to the holes shown as used to attach the guide roller carriers *G G*, but which will be found unused when guide-rollers are not applied; the tension screws *c c* placed in the extra holes that will be found in the proper place; the auxiliary springs similarly removed to the places provided for them, and the whole put together as shown in Fig. 12, if that view shows the desired direction of motion, as shown by the arrow, or as it would show if viewed in a looking-glass, if it represents the reverse of the desired direction.

The simple rule, to so put together that when the engine runs in the desired direction the pivoted ends of the levers will *lead*, the *weights follow*, and so that when the levers move outward the eccentric will be *advanced*, *i.e.*, turned on the shaft in the direction the engine is to run, will cover the case so far as instructions should be needed, the proper application of the main springs auxiliaries and guide-rollers (if any) being simply a matter of making them perform their functions as before.

The new angular position of the wheel is found by the fact that when the weight levers are on their inner stops, the governor eccentric and crank will be on their dead centers at the same time and in the same direction.

V

STRAIGHT-LINE ENGINE GOVERNOR

THIS governor, a cut of which is shown in Fig. 15, is the design of Prof. John E. Sweet. It comes under the second class of the second group described in Chapter I.

The eccentric *A* is mounted on the disk *B* and is pivoted at *C*. The eccentric center swings across the shaft center when actuated by the weight *D*. This weight is pocketed for shot to admit of changes by taking away or adding to the weight. The weight and arm are in one piece, pivoted by the pin *E*. The end of the weight-arm is connected to the eccentric disk by the link *F*. The spring *G* is made fast to the weight-arm by the band *H*. The adjustment of the spring-tension is obtained at the point *J* by slacking or screwing up the binding bolt *K*.

To increase the speed of the engine, increase the tension of the spring, or decrease the weight, or both.

To decrease the engine-speed, decrease the spring-tension, or increase the weight, or both.

Bear in mind that if the proper sensitiveness has been reached and only the speed is to be changed, the change should be made in the weight alone.

If the governor is sluggish, first see that everything

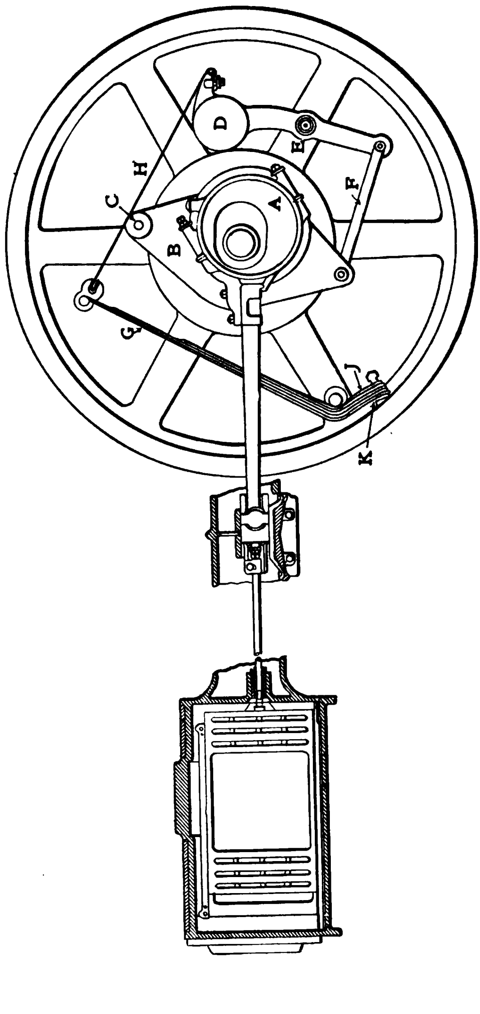

FIG. 15

relating to the valve-motion is free; then, if still sluggish, add more spring-tension and more shot in the weight pocket.

If the governor races, it may be due to sticking in some of the joints or in the valve-rod; if these are free, decrease the spring-tension and take away shot from the weight.

VI

IDEAL ENGINE GOVERNORS

THE A. L. Ide and Sons Co. use the Rites Inertia Governor on the engines they now put out, and have done so for some time past. Chapter III of this book, with the remarks here given, covers all there is to be said in reference to the adjustment of these governors.

The Ide Company has made an improvement in the Rites governor in the shape of a revolvable bronze bushing shown at *A* (Fig. 16). Owing to the fact that great wear comes on this pin, this bushing is placed there, so that a new surface can be turned to the wearing side of the pin frequently. This is done with a spanner-wrench which comes with the engine. The builders recommend that the bushing be revolved a little each day when the governor is oiled. On the face of the lug *B*, on the pulley-spoke to which the spring is attached are stamped figures which indicate, first, the speed of the engine, and second, the distance that the eye-bolt should extend through the nuts in order to adjust the governor as it was adjusted when the engine was tested in the shop. The spring is attached to the governor-bar by means of a sliding block *C* (Fig. 16). The block is in

the correct position when the line marked on it is even with the line marked on the bar.

These builders, in former years, put on the market a centrifugal governor of which Fig. 17 is a cut, and

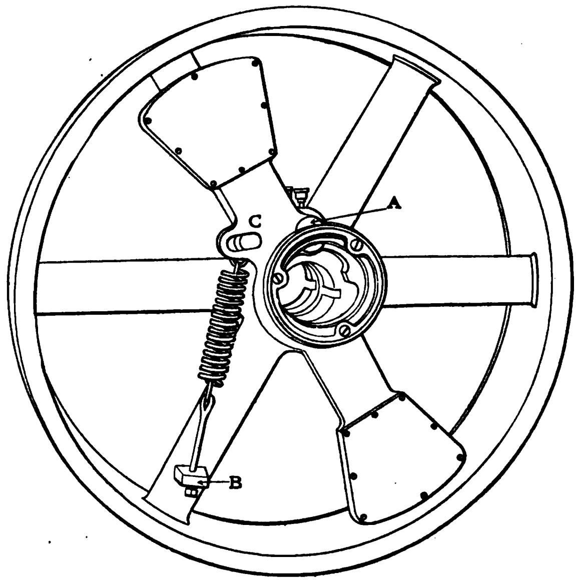

FIG. 16

as many are still in use, some instructions regarding them will follow. In taking this governor apart for oiling and cleaning, allow the sliding block *A*, which holds the end of the governor-spring, to remain with its outer edge on a line with the mark across the face of the slide, and in readjusting the spring, place the

same tension on it as was on it originally. This can be ascertained by measuring the length of thread through the nuts before slackening them. On this type of governor which is designated in the second

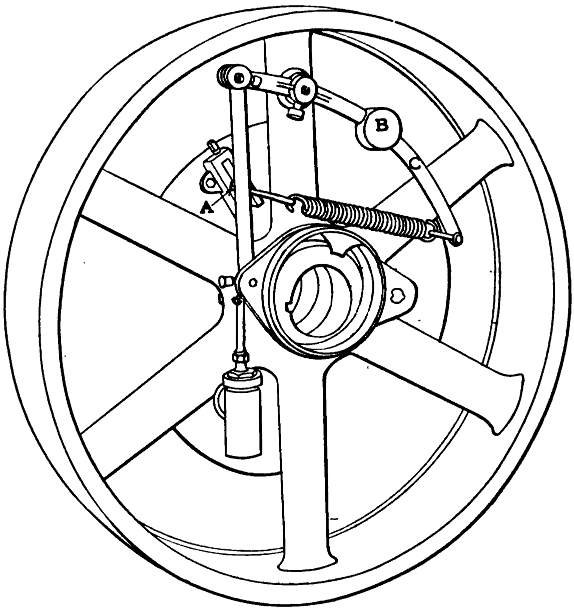

FIG. 17

class of the second group in Chapter I, the weight *B* can be moved back and forth on the lever *C* by slackening the set-screw until the weight can be moved by hand. This will have the effect of adding to or taking from the weight.

Moving the weight out toward the end of the lever has the effect of increasing it, and *moving it in* toward the fulcrum pin *D* has the effect of decreasing the same.

Changes of speed should be made with the weight.

To get increased speed move the weight in toward the fulcrum-pin.

To decrease speed, move the weight toward the end of the lever.

To make the governor more sensitive, move the block A, Fig. 17, toward rim of wheel.

To make it less sensitive and correct it for racing, move block A toward hub of wheel.

The face of the slide is marked with a line where the outer edge of the block which holds the spring should stand. Figures stamped on the face of the slide show the distance that the end of the eye-bolt should extend through nuts. This gives the right tension on the spring. Tightening the spring will give closer regulation, but if the spring is too tight, it will cause the governor to "race." "Racing" caused by over-tension of the spring can be stopped by moving the block nearer to the center of the wheel.

VII

ADJUSTMENT OF FLEMING ENGINE GOVERNORS*

The governor used on Fleming engines, built by the Harrisburg Foundry and Machine Works of Harrisburg, Pa., is of the "Centrally Balanced Centrifugal Inertia type," shown in Fig. 18. Assuming one of these governors to be out of adjustment, the weights being removed from pockets *A* and *B* and the springs loose, in order to properly adjust proceed as follows:

First Adjustment

Locate the outer ends of the springs about the center of the slots, refer to table (page 76) for the size of spring corresponding to that in the governor, noting the initial deflection. Draw up the two bolts *C*, *C*, sufficiently to stretch each of these springs by the amount of this deflection. Now start the engine and bring it up to speed, pocket-weights being removed and springs given tension shown in the same table. If the engine runs much too slowly the springs are too light and a heavier set should be used to get the desired speed. If, on the other hand, it runs too fast, add one

* This governor comes under the second class of group two in Chapter I.

weight of equal thickness to each of the pockets, *A* and *B*, placing the weights of larger diameter in *A*

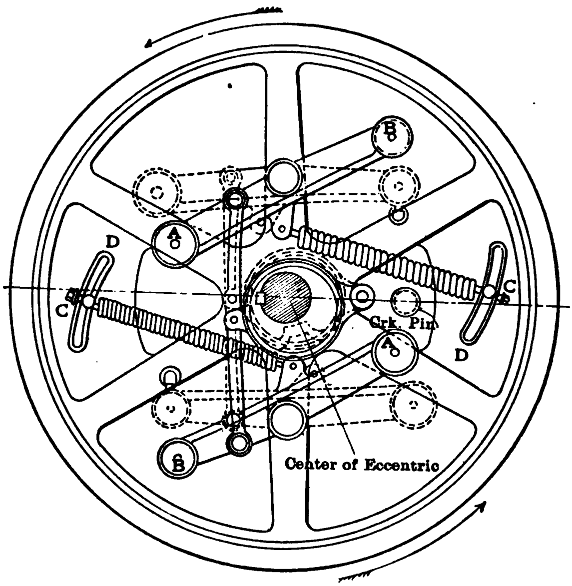

FIG. 18

pockets and the smaller ones in *B* pockets; if it still runs too fast, add another set of weights of equal thickness, selecting the proper thickness to reach the desired speed.

SPRINGS FOR HARRISBURG GOVERNORS

O. D.	Wire	Total Coils	Init. Def.	Def. Due to Gov. Throw	Total Extension
2"	5/16"	23	1 1/2"	1 1/8"	2 5/8"
1 3/4"	5/16"	27	1 9/16"	1 1/8"	2 11/16"
2"	3/8"	33	1 3/8"	1 1/2"	2 7/8"
2 1/8"	3/8"	33	1 5/16"	1 1/2"	2 13/16"
2 1/4"	3/8"	33	1 1/2"	1 1/2"	3"
2 1/4"	7/16"	33	1 9/16"	1 7/8"	3 7/16"
2 3/8"	7/16"	35	1 1/4"	1 7/8"	3 1/8"
2 1/4"	7/16"	39	2"	2 1/4"	4 1/4"
2 1/2"	1/2"	39	1 3/4"	2 3/4"	4 1/2"
2 3/4"	9/16"	39	2 3/8"	2 1/2"	4 7/8"
3 1/4"	5/8"	31	3 1/2"	2 3/4"	6 1/4"
3 1/2"	5/8"	33	3"	2 3/4"	5 3/4"

TO ADJUST TO THE PROPER POINT OF SENSITIVENESS

If the governor "races" or "weaves," move the clamp to which the outer end of the spring is attached in the slot farther from the rim of the wheel, that is, toward *D*. If this does not entirely correct the racing tendency, screw the spring-plugs farther into the springs and adjust the tension for proper speed. Taking out thin weights of equal thickness from each pocket and reducing the spring tension also assists in checking a racing tendency.

TO CORRECT SLUGGISHNESS

If the governor is too sluggish, that is, not sufficiently sensitive in order to reach the proper speed,

add a thin weight of equal thickness to each pocket and increase the spring-tension. The spring-tension, however, must not be increased to such an extent as will make the initial deflection, when added to the deflection or tension due to governor throw, greatly exceed the total deflection shown in the last column of table, and corresponding to that of these springs. While the total extension of the springs may sometimes slightly exceed that given in the table, there is danger of injury to the spring by a greater extension. If still greater sensitiveness is desired, move the clamp to which the outer end of the spring is attached, in the slot nearer to the rim of the wheel. Screwing the plugs a part of a turn out of the springs and increasing the tension will make the governor more sensitive.

If, with these adjustments, the governor cannot be made sufficiently sensitive, the springs are too heavy, and a lighter set should be used.

In cases where these governors are equipped with dash-pots, a sluggish action of the governor on starting up in a cold engine-room is sometimes due to the fluid in the dash-pot being cold and thick. This trouble will usually disappear after the engine has run a short time.

To Correct for Speeding Up

If the engine speeds up when the load is thrown off, it is either because the valve has too much lead or is sticking through lack of proper lubrication, or may possibly be leaking, due to wear, as speeding

up, due to the adjustment of these governors, is not likely to occur.

Care of Governor

The governor is a simple piece of mechanism, but it is one of the most important parts about the engine,

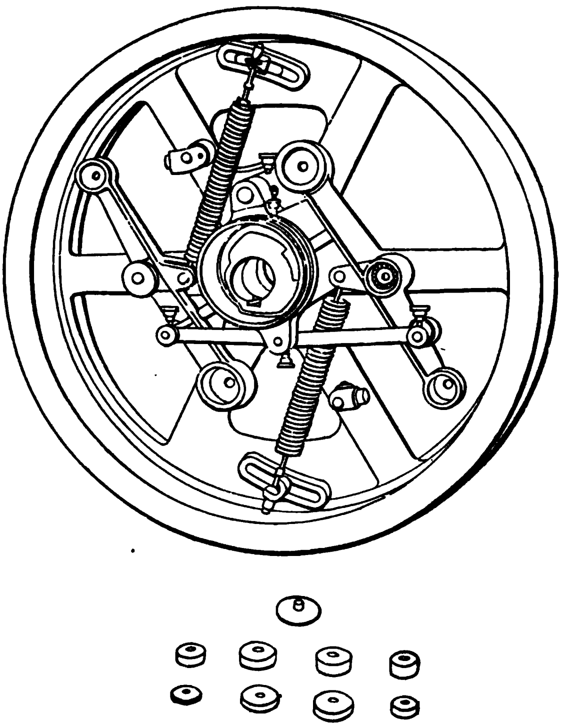

FIG. 19

and should be so treated. The springs should be disconnected occasionally, and the governor parts and valve gearing should be tested, by hand, for freedom of all bearings and joints. It is also a good plan to take the governor bearings apart occasionally, and examine them to see that they are getting proper lubrication. Clean them thoroughly before putting them together again. Before starting up the engine always see that all bolts and nuts are tight. If the governor is equipped with dash-pots keep them full of either glycerine or equal parts of cylinder and engine oil. Fig. 19 shows the governor with the weights out of the pockets.

VIII

McINTOSH, SEYMOUR AND CO.'S ENGINE GOVERNOR

THIS type of governor comes under first class and group of Chapter I. The governor is shown in detail

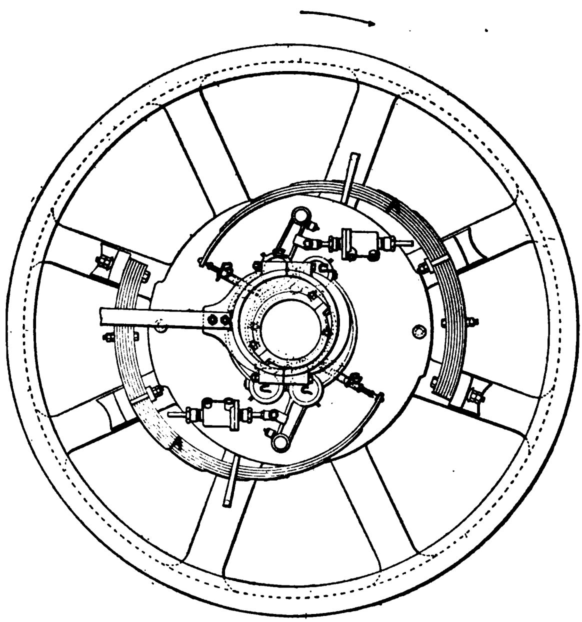

FIG. 20

in Fig. 20. This figure shows the one governor in two positions.

The position of the governor parts when the engine is not running is shown at the left. The centrifugal

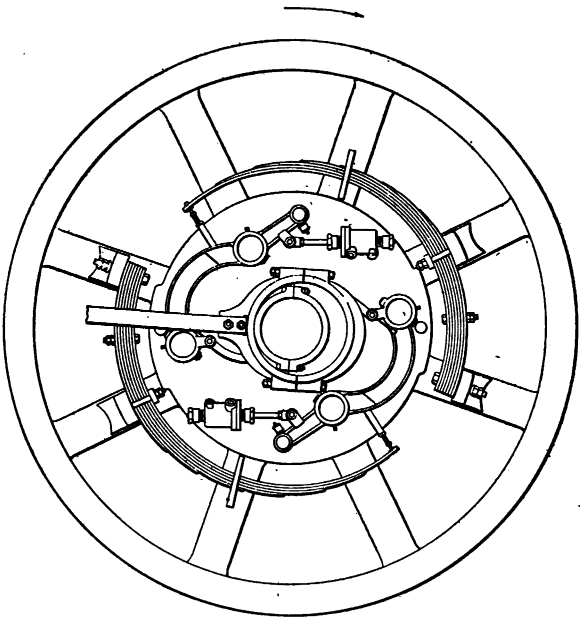

FIG. 20

weights are at their inner limit of travel and the governor eccentric is so placed as to give maximum cut-off. In the view at the right the centrifugal weights have moved into their extreme outer position, and at the same time have pulled ahead the eccentric, to

which they are connected by links and which is free to revolve on the shaft, sufficiently to cut off the steam entirely from entering the cylinder. This condition is approached when the engine is running and the load is thrown off. The centrifugal force of each weight is opposed in a direct and practically frictionless manner by a plate-spring *A*, *A*, through a hardened steel pin *B*, *B*, with a ball-and-socket bearing at the end of the spring and at the center of gravity of the weight, so that there is no friction or pressure due to this force upon the pin upon which the weight swings. This permits the use of a very heavy weight, having great centrifugal force and making the governor powerful. There are provisions for grease lubrication of all wearing surfaces. The tension pins between springs and centrifugal weights are arranged to telescope, in order that they can be adjusted to secure proper sensitiveness; for by lengthening these pins the governor can be made to regulate more closely, and by shortening them, over-sensitiveness or racing can be removed. Dash-pots are provided, which give stability to the governor, so that it can be adjusted to give nearly perfect regulation without any tendency to race under a fluctuating load.

The speed at which the engine will run can be raised or lowered by reducing or increasing respectively the small lead weights *C*, *C*, *C*, *C*, provided for that purpose in holes in the centrifugal weights. This adjustment should be made last, for it does not alter the sensitiveness of the governor to change the speed in this way,

while any adjustment of the sensitiveness as described above also changes the speed.

The governors of McIntosh and Seymour engines, when designed for driving alternating-current generators in parallel, are provided with patent compound time-delayed dash-pots, without which successful parallel operation is impossible with generators of large size and high frequency. When two alternating-current generators are running in parallel, each generator has a tendency to oscillate back and forth with reference to the other, with periodic transfer of load from one generator to the other, called "surging." A governor, which is properly sensitive, without the time-delay dash-pot, must respond to these fluctuations of speed, and when the conditions are such as exist with large generators of high frequency, resonance is produced; that is, the action of the governor tends to increase the speed fluctuations, causing the surging to build up from an imperceptible beginning until parallel running is impossible. If the governor is dampened by ordinary devices sufficiently to stop this effect, it will fail to control the speed properly, with danger of the engine running away if a considerable part of the load is suddenly thrown off. The compound time-delay dash-pots dampen heavily the governor-action for any fluctuations of speed of very short duration, such as those just described; but under the action of even the slightest change of speed, if persistent beyond this short interval of time, they automatically release the governor avoiding any impairment whatever of the speed regulation.

A speed changer is sometimes placed on governors where synchronizing of units is desired.

The mechanism of the speed changer consists of an auxiliary weight arranged to slide on the main centrifugal governor-weight, while the engine is running, in such a way as to change the speed of the engine by altering the centrifugal force to be resisted by the governor spring. The auxiliary weight is moved by a screw which in turn is rotated by a small electric motor mounted on the governor-weight.

This motor can be connected electrically, through a collector on the engine shaft, to a double-throw starting-switch on the station switchboard, in such a manner that the amount and direction of the motion of the electric motor can be controlled by the starting-switch so as to give the desired change of speed.

Adjusting Governor of a New Engine

Put all the lead pieces in the holes in governor-weights and tighten the set-screws well down into them. Then, with the shortest length of tube in the governor adjusting pins (*B*, *B*, Fig. 20), put the pins in place between ends of springs and governor-weights, care being taken to have the ends of pins well greased. Be sure that the bolting of governor-spring is secure, and that all governor parts are ready for service. Then start the engine non-condensing and without load, opening the throttle little by little so that the speed may increase very gradually.

Count the speed from time to time to make sure

that it does not exceed the rated or normal speed by more than 5 per cent. At no time should the speed be allowed to exceed this amount. If the above instructions have been followed the governor will probably control the engine at some speed considerably below its normal speed. If, however, the engine runs up above normal speed, and the governor-weights have not then opened wide, the governor does not control the speed properly and it may be necessary to change its adjustment. Before doing this, however, make an examination as follows: See that the springs do not rub hard against the spring-guides, and that they do not strike the bottom of spring-guide or any other part of the wheel when in outer position. Then remove the springs, disconnecting the auxiliary eccentrics from the valve-gear, and see that the governor-weight when connected to eccentric-sleeve, swings freely from inner to outer positions and strikes against stop-pins. Make perfectly sure that eccentric-sleeve turns freely on shaft. Connect up valve-gear again and make sure while turning the engine a complete revolution, that the cut-off valves are entirely closed when the governor-weights are in the outer position.

If no trouble has been discovered in any of these particulars remove the second leaf from each spring, considering the shortest leaf as the first. Then start the engine and run up to speed as before. If the weights do not open, remove the fourth leaf, and, if necessary, the fifth and sixth.

The object of the foregoing operations is to secure

governor-control of the engine at some speed below the normal, and at the same time obtain a sluggish regulation. The next step should be to secure correct adjustment of the sensitiveness of governor, to give proper closeness of regulation, after which the speed should be adjusted to the desired number of revolutions per minute.

When the governor-control has been secured as above, the sensitiveness of governor will probably be found to need increasing by increasing the length of the adjusting-pins between the governor-weights and the ends of the governor-springs. The adjusting-pins should be gradually lengthened one-half inch at a time, until the proper sensitiveness is reached, always keeping the length of the pins the same. These adjusting-pins should have been unscrewed before putting them in position, and the length of the threaded parts measured, as, when in position, at least 1¼ inches of threaded part must always be left in the socket. If longer pins are required than this will allow, put in the next longer set of the tubular parts of the adjusting-pins.

At the start the sensitiveness of the governor should be made such that when the load is removed the increase of speeed will be not less than 3 per cent. After the engine has run awhile the sensitiveness can be increased sufficiently to make the corresponding increase 2 per cent. In determining the speed of an engine always count the speed for several consecutive minutes, and divide the total number of revolutions by the number of minutes during which the speed is

counted. The speed-light should always be taken after the load has been removed.

In many cases it is not convenient to secure a load for testing the sensitiveness of governor, as has been just described, when an engine is first started, and generally the easiest way of securing a proper preliminary adjustment of sensitiveness, with engines of small size, is to continue lengthening the adjusting-pins until the engine "races." Then reduce the length of pins until "racing" ceases. With large engines it is frequently impossible to make them race. In such cases an approximate preliminary adjustment of sensitiveness may be made, when not convenient to secure a load for engine, by lengthening the adjusting pins until speed of engine is from 5 to 10 per cent. below normal.

After securing a more or less perfect adjustment of sensitiveness of the governor, as above, bring the engine up to speed by reducing the amount of lead in the holes in the governor, or the centrifugal weights. Begin by removing one-half the lead from the hole in each governor-weight which is farthest from the pin on which it turns, replacing the lead removed with a similarly shaped piece of hard wood to secure the remaining lead. The resulting change in speed of engine will give an approximate idea of how much should be removed to secure the desired speed, bearing in mind that removing lead from the holes in weights farthest removed from the pins on which weights turn will affect the speed three or four times as much as will a similar change in holes nearest these pins,

and that the same amount of lead should be kept in corresponding holes in each weight. It is intended that the engine should regulate well and be at proper speed with every hole in the weights about one-half filled with lead, but the effective stiffness of springs is quite uncertain, and the necessary amount of lead will vary to correspond. If, with all lead weight out, the speed is still too low with the governor sufficiently sensitive, one or more leaves must be added to each governor-spring, placing the added leaves between the longest leaf and the leaf next to it in each case.

Fundamental Principles for Regulating a Governor

To make a governor more sensitive, increase the tension in springs by lengthening the adjusting pins; to make it less sensitive, reduce the tension by shortening the pins. To increase speed of engine, remove lead weights from governor-weights; to decrease speed, increase the amount of lead in the weights.

These two principles should be studied carefully until thoroughly understood, as nearly all failures to successfully regulate a governor are caused by disregarding them.

In this connection, always remember that altering the sensitiveness by changing the length of adjusting-pins, also alters the speed of the engine. The speed should be brought back to that desired by a proper change in the amount of lead in the weights. Chang-

ing the speed by changing the amount of lead weights practically does not affect the sensitiveness.

Delay Dash-Pots

Engines designed for operating directly connected alternators in parallel are provided with patent delay dash-pots; otherwise the alternators will, under certain conditions, set up periodic cross-currents which may keep increasing in strength until the units are forced out of step. The delay dash-pot prevents this "surging" of currents with any generator not abnormally sensitive, but at the same time does not affect the regulation of the governor for actual change of load. As it is of the greatest importance to the proper action of these dash-pots that they be kept completely filled with oil, they should be filled every time the engine is shut down.

Directions for adjusting these patent delay dash-pots should be secured, if necessary, from the builders of each particular engine.

Operators often request information of the builders in reference to their left-hand or right-hand governors. The builders need some information from the operators before giving full instructions, and Figs. 21 and 22 show a cut of the data sheets they desire to have filled out. A study of these will enable the operator to give the data desired without first sending to the shops for such sheets.

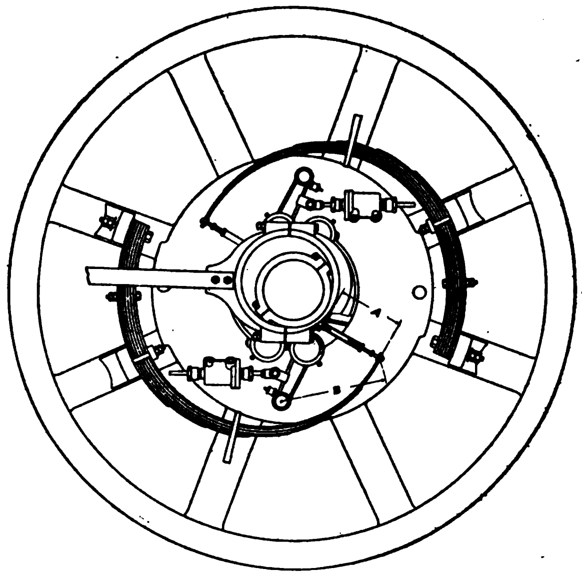

FIG. 21

SPEED....Revs. with no load. ‖...Revs. with...H. P. ‖...Revs. with....K.W.

PLAIN WEIGHT-ARM. SPEED CHANGER WEIGHT-ARM.

Sliding weight in....position.

No. of leaves in spring,.... No. of leaves in spring,.....

Lead weight in inner hole,.... in outer hole,.... Lead weight in outer hole, ...

*A............†B.......................... *A........†B.............

Remarks:...

....

....

Signed,..

Date...

NOTE—Plain governor weight-arm has two holes which hold lead weights for adjustment of speed. Outer hole is the one farthest from pin on which arm is pivoted. Speed changer governor weight-arm has no inner hole. In giving amount of lead weight in hole, state what proportion of hole is filled with lead, *i. e.*, "half full," "quarter full," etc. FILL OUT THIS REPORT AND RETURN TO SHOP AS SOON AS GOVERNOR IS ADJUSTED SATISFACTORILY.

*A = Length over all of adjusting-pin.

†B = Distance from center of weight pin to center of spring cup.

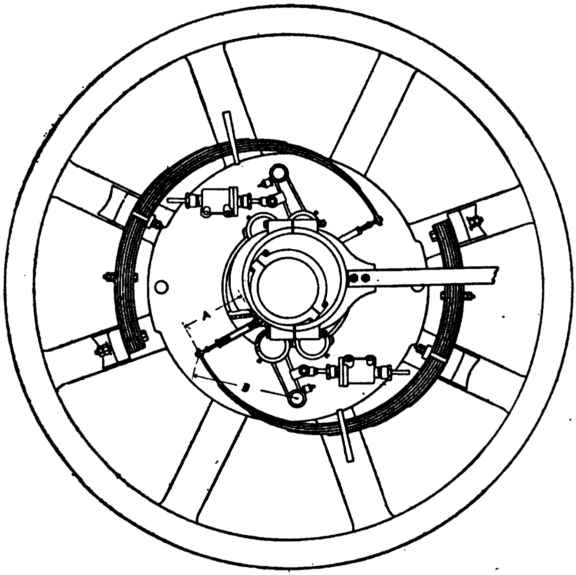

FIG. 22

SPEED . . . Revs. with no load. ‖ . . . Revs. with . . . H. P. ‖ . . . Revs. with . . . . K. W.

PLAIN WEIGHT-ARM.

No. of leaves in spring, ...
Lead weight in inner hole, ... in outer hole, ...
*A . . . . †B . . . .

SPEED CHANGER WEIGHT-ARM.

Sliding weight in . . . . position.
No. of leaves in spring, ...
Lead weight in outer hole . . . .
*A . . . . †B . . . .

Remarks: . . . .
. . . .
. . . .
Signed . . . .
Date . . . .

NOTE.—Plain governor weight-arm has two holes which hold lead weights for adjustment of speed. Outer hole is the one farthest from pin on which arm is pivoted. Speed changer governor weight-arm has no inner hole. In giving amount of lead weight in hole, state what proportion of hole is filled with lead, *i. e.*, "half full," "quarter full," etc. FILL OUT THIS REPORT AND RETURN TO SHOP AS SOON AS GOVERNOR IS ADJUSTED SATISFACTORILY.

*A = Length over all of adjusting-pin.

†B = Distance from center of weight pin to center of spring cup.

IX

ROBB-ARMSTRONG-SWEET GOVERNOR

A CUT of the governor manufactured by the Ames Iron Works for use on their engines is shown in Fig. 23. This governor is placed in the second class and group

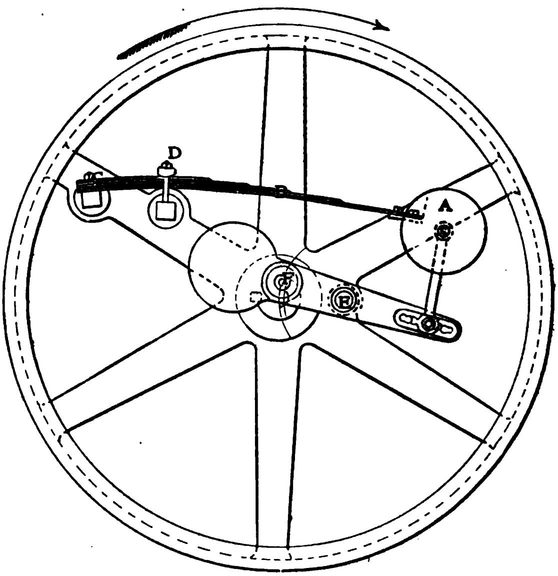

FIG. 23

in Chapter I. The weight *A* is fastened directly to the spring *B*, which is secured at *C*. The tension on the spring is changed by taking up or slackening the tension-studs *D*. The eccentric-arm is pivoted at *E*, moving the eccentric-pin *F*, which changes travel of valve and point of cut-off. The arm is actuated by the spring direct, by means of the one link *F*, one end of which can be changed in its position by shifting the pin into any one of the series of holes shown.

To increase speed, give more tension on the spring.

To decrease speed, give less tension on the spring.

To get closer regulation, and more sensitiveness, move the pin in the eccentric lever closer to the shaft-center.

To make more sluggish and put a stop to racing, move the pin in the lever toward the rim of the wheel.

No change of weight is provided for, as the above allowance for change is considered by the makers to be sufficient to cover all requirements.

X

THE FITCHBURG STEAM-ENGINE GOVERNOR

THE type of governor shown in Fig. 24 is in the second class of the first group of Chapter I, and is of the patent and manufacture of the Fitchburg Steam-Engine Company, used on all engines of their make.

The small weights shown are to counterbalance the weight of valves, stems and eccentric, and are not to be considered in the adjustment of the governor. The weights *A, A* are changeable. *Adding weight* decreases speed, and *taking it away* increases it. The weight-arms are pivoted at *B, B*, and are opposed by the springs *C, C*, which are attached, as shown, directly to the weights.

Tightening the springs, increases speed and sensitiveness.

Slackening springs, decreases speed and sensitiveness.

These engines are so carefully adjusted in the shops as to require little change of weight. The principal changes for speed and sensitiveness are to be made on the springs.

To get more speed, tighten the springs.

To lessen the speed, slack off on springs.

To get more sensitiveness, increase tension on springs;

or, if speed is already attained, increase the tension and weight at the same time to keep the speed at the same point.

To make more sluggish, decrease spring-tension, and

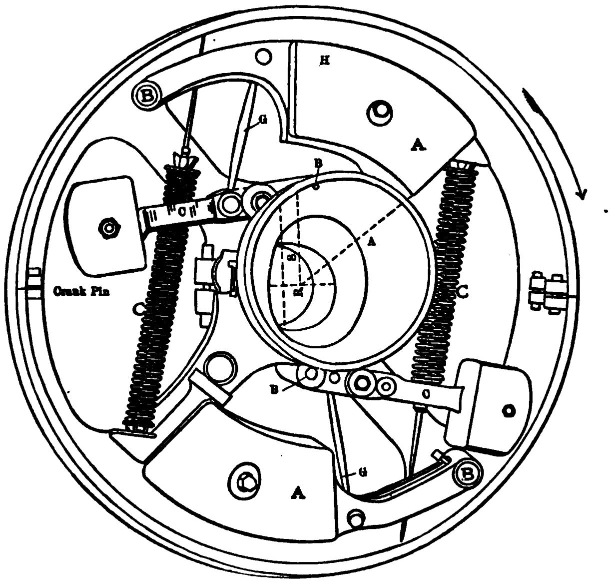

FIG. 24

if speed is right, decrease weight also to keep the speed at the same point.

As correct valve setting is necessary to good regulation, the following extract from the builders' instructions as to how to locate the governor-case on the shaft will be of service.

The location of the governor-case is determined by placing the engine on one dead center and rolling the case around the shaft until the offset of the eccentric is on the opposite side of the shaft from the crank-pin. Then roll carefully into such position that when (with the springs removed) the eccentric is thrown back and forth across the shaft, no end motion is given the valve-rod. At this place tighten the governor-case firmly upon the shaft and turn the engine to the opposite dead center, and again move the eccentric back and forth across the shaft. If there is at this end any end motion to the valve-rod, change the position of the governor-case on the shaft enough to make the motion just half as much, then fasten the governor-case firmly in this final position by drilling into the shaft for the point of the set-screw and then tightening the clamp-bolts to place solidly. Put in the springs and tighten them until the proper number of revolutions is obtained. Be sure to tighten up those that go through the counterbalance which hangs nearest the springs (when the governor is at rest) about three-fourths of an inch more than the springs on the other side.

When it is desired to change the direction of rotation of a Fitchburg engine a new eccentric must be procured from the makers and put on in place of the one on the governor.

The ends of the links which connect the weight-arms must be changed, on the counterbalance weight-arm end, to the holes opposite those which they occupied in the old eccentric.

XI

THE AMERICAN-BALL BALANCED AUTOMATIC GOVERNOR

HEREWITH is illustrated a new type of fly-wheel governor, manufactured by the American Engine Company, of Bound Brook, N. J., and with which the American-Ball engines are now equipped. It is the outcome of redesigning the Ball balanced automatic governor.

In the new type, Fig 25, two features are embodied, one being the method of establishing a gravity balance, and the other the arrangement and relation of the springs, of which there are two. A second arm is provided in the governor, as shown in Figs. 26 and 27, which is so pivoted that its center of gravity practically coincides with the center of the shaft, and therefore cannot develop centrifugal force. The arm *B* is pivoted at the most desirable point for determining the path of motion of the valve-actuating pin, the second arm *B* being so connected to the centrifugal governor that the gravity of one is always opposed by the gravity of the other at every position of the governor-wheel. By this arrangement the centrifugal force of the governing-arm, under the control of the spring, governs the engine, and the disturbing

gravitation of the arm is balanced by the opposing gravity of the second arm, which has practically no centrifugal force.

Attention is especially directed to the arrangement of the double springs for the prevention of the trouble-

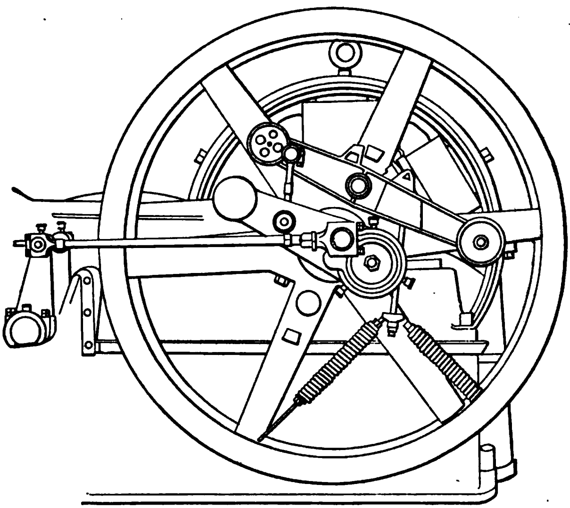

FIG. 25

some swaying characteristic of a single spring, when used, due to the centrifugal force and gravitation. These springs are convenient for slight adjustments for the difference in speed at the several points of cut-off.

Should the speed decrease under load more than is desirable, this fault may be corrected by slacking the

spring C and tightening the spring D which makes the governor more nearly isochronous. On the other hand, if the action of the governor is unstable, slacking the spring D and tightening the spring C will correct it.

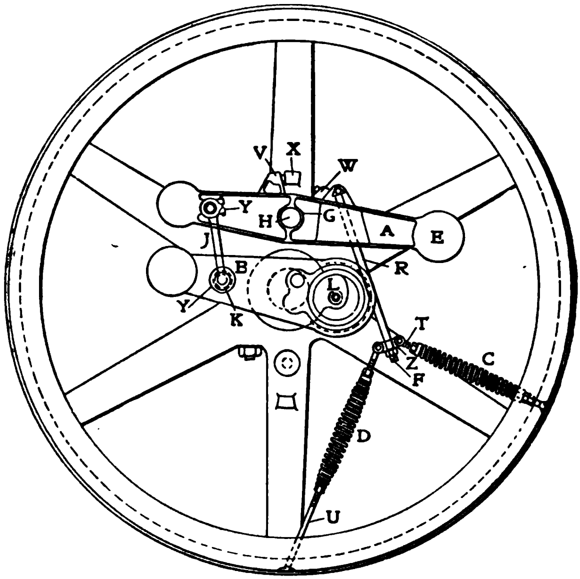

FIG. 26

For slight changes of speed, the nut F may be tightened or slacked, but for a considerable change of speed it is necessary to add to or take from the weight in the pocket E of arm A.

In Figs. 28 and 29 are shown the parts of which the governor is composed. It will be seen that the gov-

ernor-weight or arm *A* is provided with a brass bushing *G*, in which three oil grooves are cut which permit of freely lubricating the steel-governor weight-stud *H*. The arm *A* is connected to the eccentric carrier arm *B*

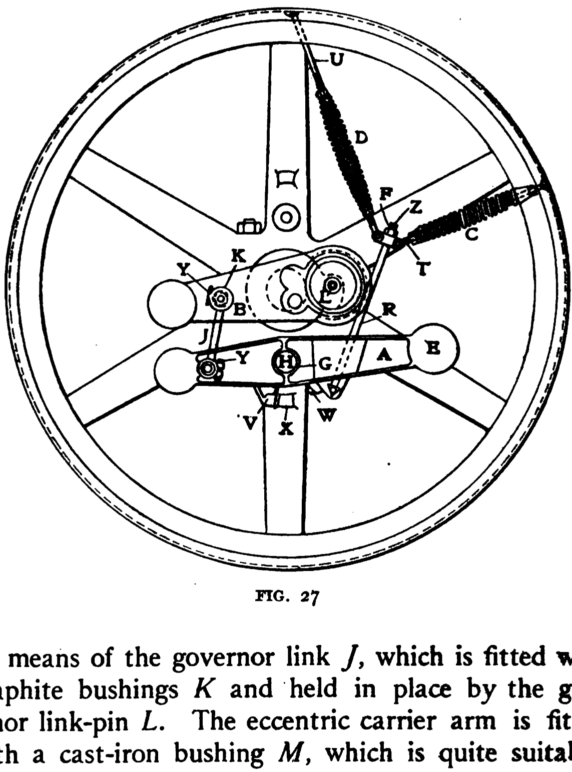

FIG. 27

by means of the governor link *J*, which is fitted with graphite bushings *K* and held in place by the governor link-pin *L*. The eccentric carrier arm is fitted with a cast-iron bushing *M*, which is quite suitable, there being so little movement at this point that a bushing of special material is unnecessary. At the

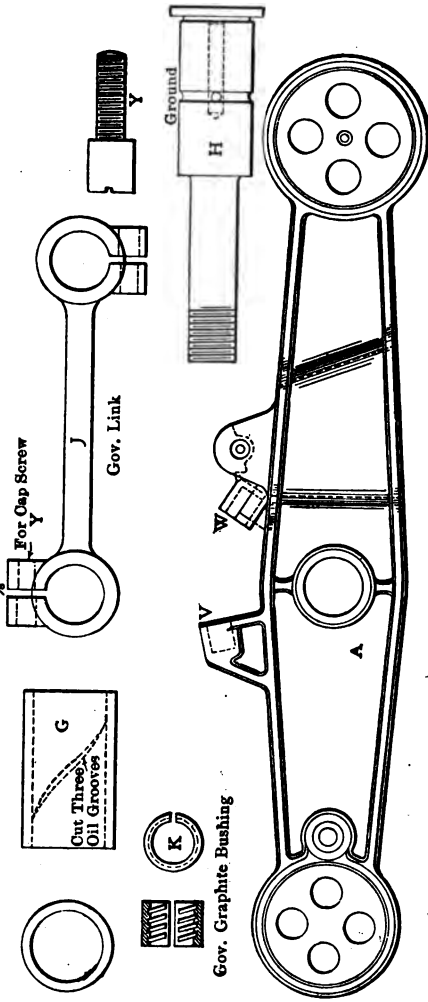
Y
Ground
H
For Cap Screw
Y
⅝"
J
Gov. Link
W
V
A
G
Cut Three
Oil Grooves
K
Gov. Graphite Bushing

FIG. 28

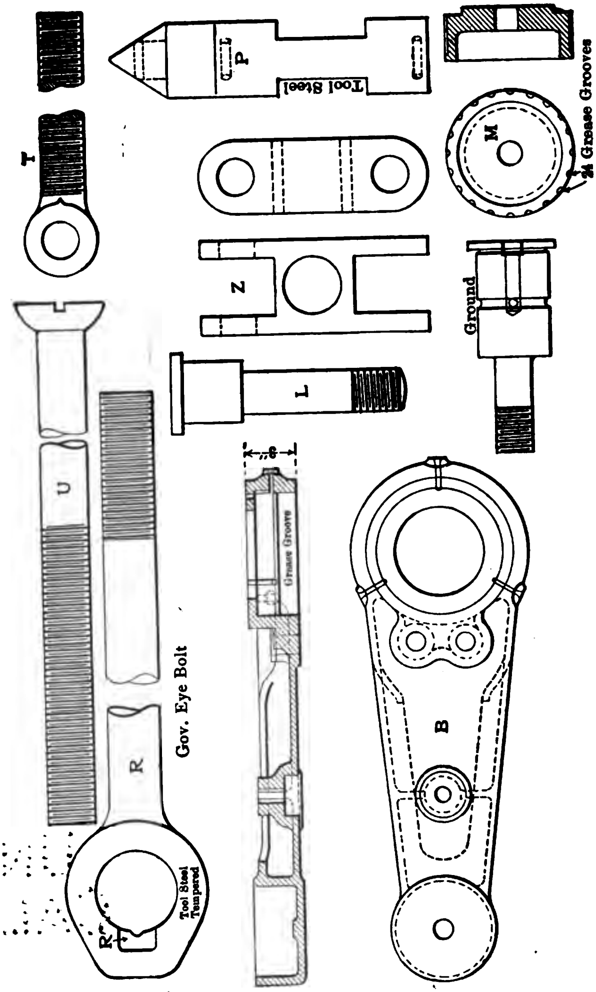
T
P
Tool Steel
M
Grease Grooves
Z
Ground
L
U
R
Gov. Eye Bolt
Tool Steel Tempered
Grease Groove
B

FIG. 29

point *O* in the boss on the arm *A*, a tempered knife-edge *P* is inserted. Three notches are filed in the hole so that the knife-edge will fit snugly and not turn. On this knife-edge is suspended the governor-spring eye-bolt *R*, in the eye of which is fitted a piece of tempered tool-steel at R^1, which wears on the tempered knife-edge. This eye-bolt is threaded at the opposite end, over which is fitted the governor-screw spring-clip *Z*, which is held in place by a nut and lock-nut. The springs *C* and *D*, Figs. 26 and 27, are screwed into the spring-eyelet *T* at one end and the spring-screw *U* at the other.

The arm *A* has two lugs cast on it at *V* and *W*, in which are fitted a piece of round fiber, which, coming in contact with the lug *X* on the governor-wheel, fixes a limit to the movement of the arm *A*.

These governors are made for engines running over, unless ordered otherwise, although provisions have been made for permitting of changing to governors running in the opposite direction. If, for instance, an engine were equipped with a right-hand governor so that it ran over, and it was desired to operate the engine in the opposite direction, it would be necessary to drill holes for the arrangement of the proper pins and springs as shown in Fig. 27. The position of the governor would then become reversed and the engine would operate in the reversed direction.

XII

CURTIS STEAM TURBINE GOVERNORS

THE General Electric Company, in the manufacture of the Curtis Turbine, uses a governor of the spring-loaded fly-ball type on the main shaft, and necessarily operating at the same speed without the introduction of intermediaries. The movement of this governor actuates the device controlling the valves admitting the steam to the turbine. The assembly of these turbines with the governor at 17 and the valves it controls at 18 is shown in Fig. 30. A detail view of this governor is shown in Fig. 31. A certain percentage of the spring effect is carried in a small spring under the control of a motor operated from the switch-board, for the purpose of varying the speed of the turbine in order to synchronize with other machines.

Referring to this figure the following is a list of the various parts of a

MAINE TURBINE GOVERNOR

1. Governor bracket.
2. Stud for frame.
3. Middle plate.
4. Top plate.
5. Nut for lower end of stud — with lock washer.
6. Nut for upper end of stud — with lock washer.
7. Strap for studs.
8. Bolt for strap — with nut and locker washer.
9. Fulcrum block.

10. Guide roller block.
11. Bolt for fulcrum and roller blocks — with nut and lock washer.
12. Guide roller — with pin and cotters.
13. Governor weight.
14. Knife-edge block — with screws.
15. Hook — with screws.
16. Plug for balance pocket.
17. Yoke for links.
18. Links.
19. Universal joint.
20. Lower governor plug.
21. Upper governor plug.
22. Governor spring.
23. Key for upper plug with screws.
24. Adjusting nut for upper plug.
25. Connection rod.
26. Gimball transmission bearing.
27. Ball races for Gimball bearings, upper and lower.
28. Gimball pivot — for box.
29. Gimball pivot — for beam.
30. Bushing for pivots.
31. Gimball ring.
32. Beam.
33. Dome — with bolts.
34. Cover plate for dome — with cap screws.
35. Bearing bracket for dome — with bolts.
36. Spindle for roller bearing.
37. Rollers for bearing. (Number.)
38. Bushing for bearing.
39. Pin for attaching synchronizing connection to beam.
40. Connection for synchronizing spring.
41. Upper plug for synchronizing spring.
42. Synchronizing spring (give dia. spring, dia. wire, active turns).
43. Traveling nut for synchronizing spring.
44. Limit switch.
45. Synchronizing motor (Series d. c. — Give rating).
46. Worm for synchronizing gear.
47. Bracket for worm.
48. Worm wheel.
49. Cap for synchronizing screw.
50. Synchronizing screw
51. Bracket for synchronizing gear — with bolts.

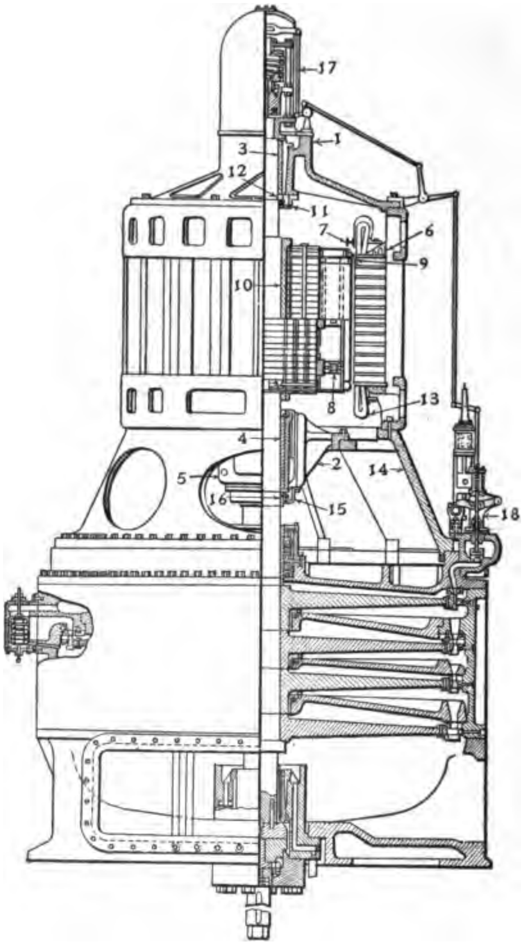

FIG. 30

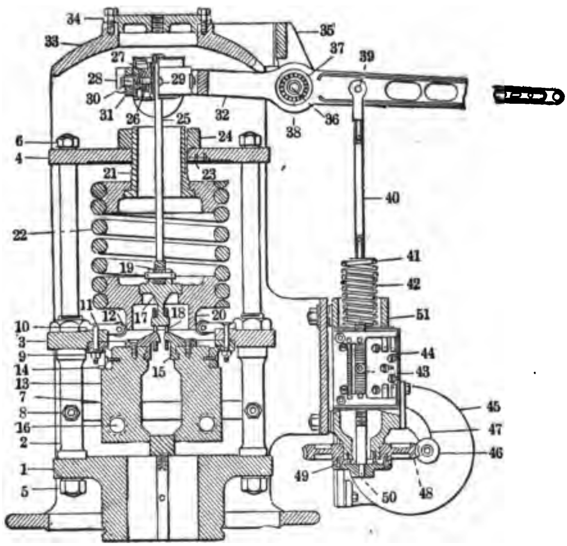

FIG. 31

Operation of Governor Explained

By referring to Fig. 32 the following explanation of the governor-action will be made plain.

The governor-bracket, holding the weights and spring, revolves with them and the shaft. The shaft extends up through the bracket at *H*. The spindle *C* revolves with the bracket and swivels in the end of the beam, which is stationary. The motion of this beam is transmitted through the rod *D* (Fig. 33) to the arm *G* and to the pilot valve of the oil cylinder *B*, containing the piston *A*, which actuates the main arm. The

main arm transmits the motion, either by means of a rack connecting with a pinion or by means of cranks, to the rod carrying the cams. These cams act directly

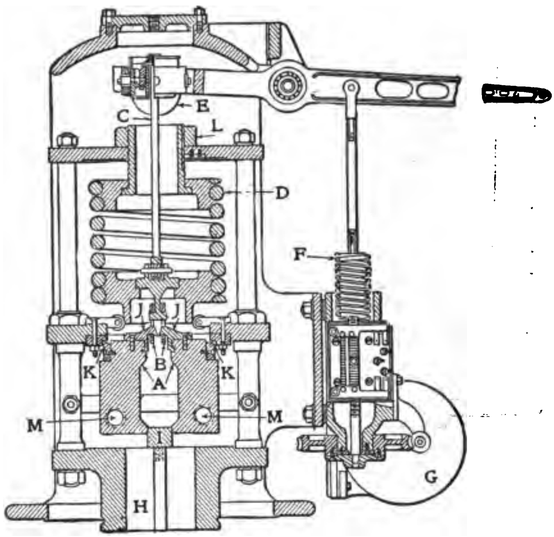

FIG. 32

on the valves, opening and closing the number called for by the condition of the load.

In Fig. 32 the governor is shown at rest, in position for full admission of steam to the turbine. The weight rests on the stop *I*, which corresponds to the inner stop of the weights of a shaft governor. The weights are fastened over a knife-edge to the links at *J*, *J*, and have their fulcrum over the edges *K*, *K*. The links hold to the yoke in the bottom of the spring,

and the other end of the spring is fastened to the top plate by means of the plug and adjusting nut. The weights act centrifugally, and as they fly out from

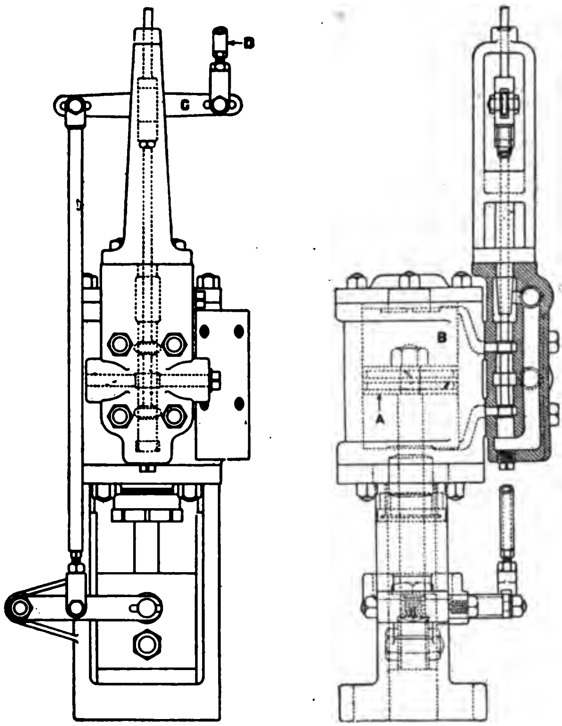

FIG. 33

the center they push against the edges *K*, *K*, and pull against the edges *J*, *J*.

With this governor as with shaft-spring governors,

tightening the spring increases speed, and slackening it, decreases the speed. To tighten the spring of this governor screw down on the adjustment nut *L* (Fig. 32), to slacken the spring, slack off on the nut.

To increase the sensitiveness or decrease the regulation of this governor, increase the number of working coils in the main spring, keeping initial tension the same.

To make the governor less sensitive, or increase the regulation, decrease the number of working coils in the main spring.

For the purpose of changing the regulation through a small range, the weights are provided with pockets for loading. Increasing the weight decreases the regulation and vice versa. Any change in the weight requires a corresponding change in the initial tension of the main spring in order to maintain the proper speed.

XIII

CHANGING THE SPEED OF PENDULUM GOVERNORS*

An old engine was brought to a machine-shop to be thoroughly repaired. When it was nearly ready to set up the question of its future speed was presented, and it was decided to run it 65 revolutions per minute. An engineer who had had charge of this engine several years before was consulted, and he reported that its former speed was 75 revolutions per minute. From this fact, in connection with measurements made to determine the diameter of pulleys used to drive it, the speed of the governor was calculated, and as all men in charge of plants do not understand the principles involved in this and similar problems, an explanation of the same will be given in a practical way.

A governor, as used to regulate the ordinary Corliss, or any similar type of engine, is illustrated in Fig. 34. In the case already referred to, the crank-shaft revolved 75 times per minute, and the pulley on it is 9 ins. in diameter (see 2 in the cut). The governor pulley 3 is 12 in. The speed of governor is $75 \times 9 \div 12 = 56$ revolutions per minute.

On some of the governors furnished to users the

* Contributed to *Power* by W. H. Wakeman.

speed is stamped, which is a great convenience; otherwise, it is necessary to determine experimentally the speed required to elevate the balls to their working plane.

The working engineer is often confused in regard to changing the speed of engines, because he fails to fix in his mind the fact that when the speed of a governor

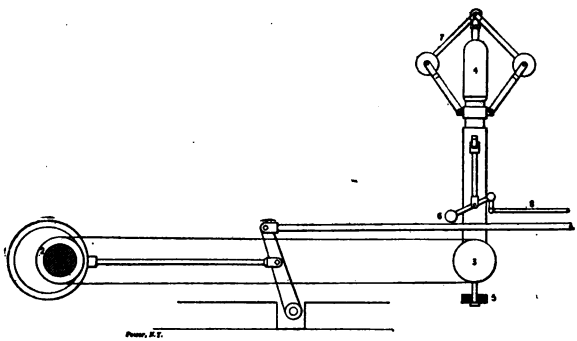

FIG. 34

is once fixed it remains unchanged, regardless of any change made in the size of pulleys used to drive it.

In a swinging-pendulum governor the centrifugal force and gravity are equal at one point only in its operation. The force of gravity is represented by the weight of the balls, and when they revolve fast enough for the centrifugal force to equal the weight the two forces are equal. The point where the two forces are equal, or nearly so, is fixed, so that when the balls are raised to the working plane by centrifugal force the governor mechanism is cutting off the steam

at its minimum point. For that reason the same speed of governor must be maintained as an increase or decrease of engine-speed hastens or delays the cut-off action beyond the proper point. If driven a little too fast, it reaches its highest plane and shuts off steam altogether; if a little too slow, it falls to its lowest plane, admitting the maximum quantity. If extra weights are added to or taken from the governor, if the tension of a spring is increased, or decreased, or the reach-rods on a Corliss engine are changed, the speed at which a governor must be driven to be kept within its operative plane will be affected, but this belongs to another part of a subject that will receive attention later.

The governor referred to revolves 56 times per minute and it is desired to run the crank-shaft 65 revolutions in the same time. Multiplying the speed of crank-shaft by the diameter of pulley 2 and dividing by the speed of governor shows that the pulley 3 should be $65 \times 9 \div 56 = 10.4$ in. in diameter.

Where the pulley 3 is to be retained and a smaller one put on the crank-shaft, the speed of governor is to be multiplied by the diameter of pulley and the product divided by the speed of crank-shaft. Then $56 \times 12 \div 65 = 10.3$ in.

Where a governor is driven by gears the same principle is involved, but some engineers do not understand it so, therefore an illustration will be given.

Figure 35 shows a governor driven from the crank-shaft by gears. Here 2 represents a gear on the crank-shaft, which drives another gear 3 on an independent stud. The latter is twice as large as the

former and the bevel gears 4 and 4 are alike, therefore the side shaft 5 makes one revolution while the crank-shaft gear 2 revolves twice.

The first two years that this engine was used it revovled 50 times per minute. The bevel-gear at 6 has

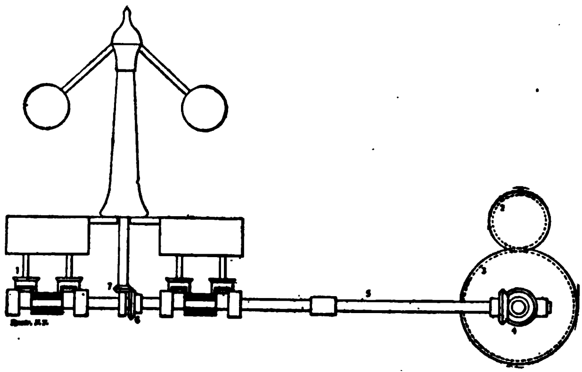

FIG. 35

44 teeth and 7 has 20, therefore the speed of governor is 25 x 44 ÷ 20 = 55 per minute.

Suppose, for example, that the 20 gear at 7 be taken off and a 30 gear be put in its place, how fast will the governor run? Some may figure it at 25 x 44 ÷ 30 = 36.7 times per minute. It has been done so, yet it is not correct. The speed of the governor remains constant; it is the speed of the engine which may be changed.

This governor revolves 55 times per minute; the new gear at 7 has 30 teeth, and 6 has 44, therefore the speed of side shaft 5 is 55 x 30 ÷ 44 = 37.5 revolu-

tions per minute. While 5 makes one turn 2 revolves twice, therefore the speed of engine should be 37.5 x 2 = 75 revolutions per minute. If this reasoning is correct (and as a careful count of the speed shows it to be 75) it proves the theory to be right.

Other means adopted for changing the speed of engines require a passing notice in order to cover the subject. If the center-weight 4, in Fig. 34, is made lighter it will decrease the speed of both engine and governor, and if made heavier it will increase the same, because it will change the plane in which the balls travel for a given speed. Some governors have hollow center-weights, so that shot can be put in or taken out at pleasure. Any change in the weights at 5 will have the same effect, as the rod which supports them is a continuation of the spindle and collar which carries 4.

This is a very convenient plan for use in connection with a governor that does not respond quickly to changes in the load; for, when a heavy machine is started up, another weight may be added at 5, and when said machine is stopped the weight may be removed. This is a crude plan when compared with modern regulating devices, but it has been found to be much better than none.

The disk 6 is on a lever, and as it is moved nearer to or farther from the fulcrum it changes the speed slightly. Some governors are adjustable at 7, so that by changing the length of arm at this point, the speed is changed. The reach-rod 8 may be made longer or shorter, thus making small changes in the speed; but neither this nor the plan just preceding it is recom-

mended, as they are not founded on desirable principles, and bring objectionable features into the matter which it is well to avoid. When a governor with its connection is properly set up, it is not advisable to change either 7 or 8, for changes in the former may affect the sensitiveness of the mechanism, and careless adjustment of either may prevent a very short cut-off, and thus cause trouble in case all of the load is suddenly thrown off.

Some Causes of Trouble with This type of Governor

In almost all makes of these governors there is a pin on which the weights are brought to rest when the mechanism is not in action. This is a safety-pin, or sometimes a collar, which prevents the mechanism from falling so low that no steam will be admitted. This pin, or collar, is so placed that when the engine is at rest it will get steam. When the engine is in full operation the pin is removed or the collar so turned that, should the belt or gear break, the mechanism would drop so low as to cut off all steam and a shut-down results.

In plants where heavy and changing loads are handled, it is not uncommon for one to come on so heavy as to make the mechanism drop low enough to shut off steam, if the operator has attended to his duty of removing the pin or setting the safety collar after starting up. The result is a shut-down, and it may confuse the inexperienced operator till the lesson is

learned and he knows the cause. Always look at the "safety" when a shut-down occurs out of the usual time.

Some governor pulleys are secured to the shaft with a set-screw which may come loose, or a key even may work loose. The pulley may hold just enough to slowly rotate the governor but not fast enough to bring it up to speed. The result will be a runaway engine. An oily or slack governor-belt may also cause this.

The following experience illustrates another cause of trouble with governors.

On a 14 x 36 Corliss engine of from 90 to 140 H. P., an overload would cause the steam to follow full stroke, as the steam-valves would not trip and cut-off. The governor, after going down until the tripping cams did not touch and trip the latches, would have a hard struggle to rise again to a point where the tripping would recommence. It seemed that the force required to trip the latches was so great that the engine speed necessary to give the governor the needed power had to be greatly accelerated, and in going through this part of the performance the governor would dance violently with every movement of the trip-rods. These conditions produced racing, or rather, "hunting."

The latches, or hook-plates, had a catch surface of $\frac{1}{8}$ of an inch and tripped very stiffly. Thicker leathers were placed in the hooks, so they did not overlap the plate on the valve-crank so far, reducing the catch surface to $\frac{1}{16}$ of an inch. At present the governors are doing their work satisfactorily, but during two and a

half years the corners have been worn completely off the latches and blocks five times. Of course this is due to the very small amount of catch surface allowed. The blocks and latches are as hard as any, but the decreased area of contact, with increased pressure on the plates, causes the increased wear. This is the sacrifice necessary to get earlier cut-off and greater steam economy. This is a case where the strain on catch-blocks must be reduced to assist the governor in its work.

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